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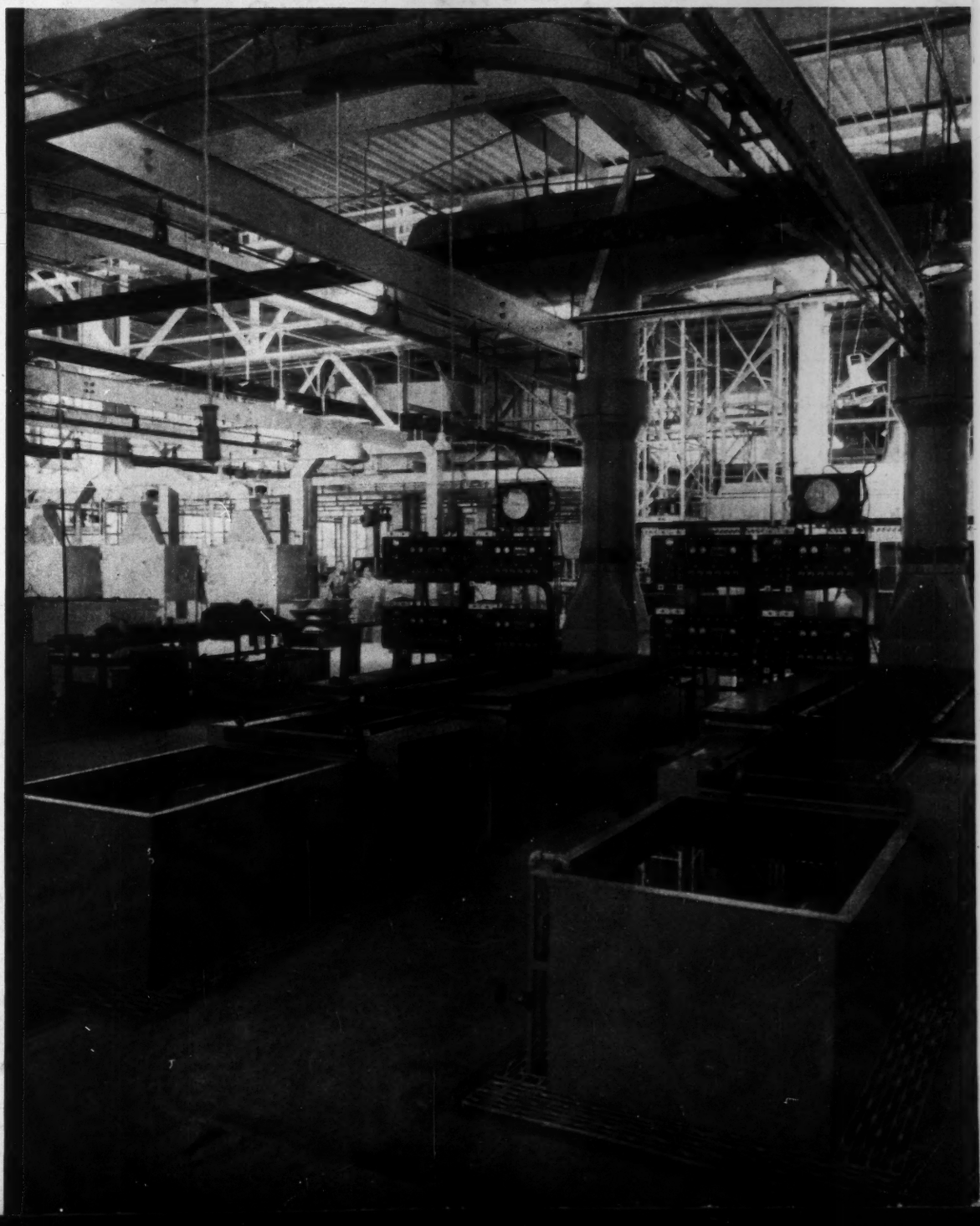

DEFENSE PROGRAM—New information for design of ventilating systems for electroplating processes used in production of defense material is given in our lead-off article this month. Illustrated in the photo is a plating department at Wright Aeronautical Corp.

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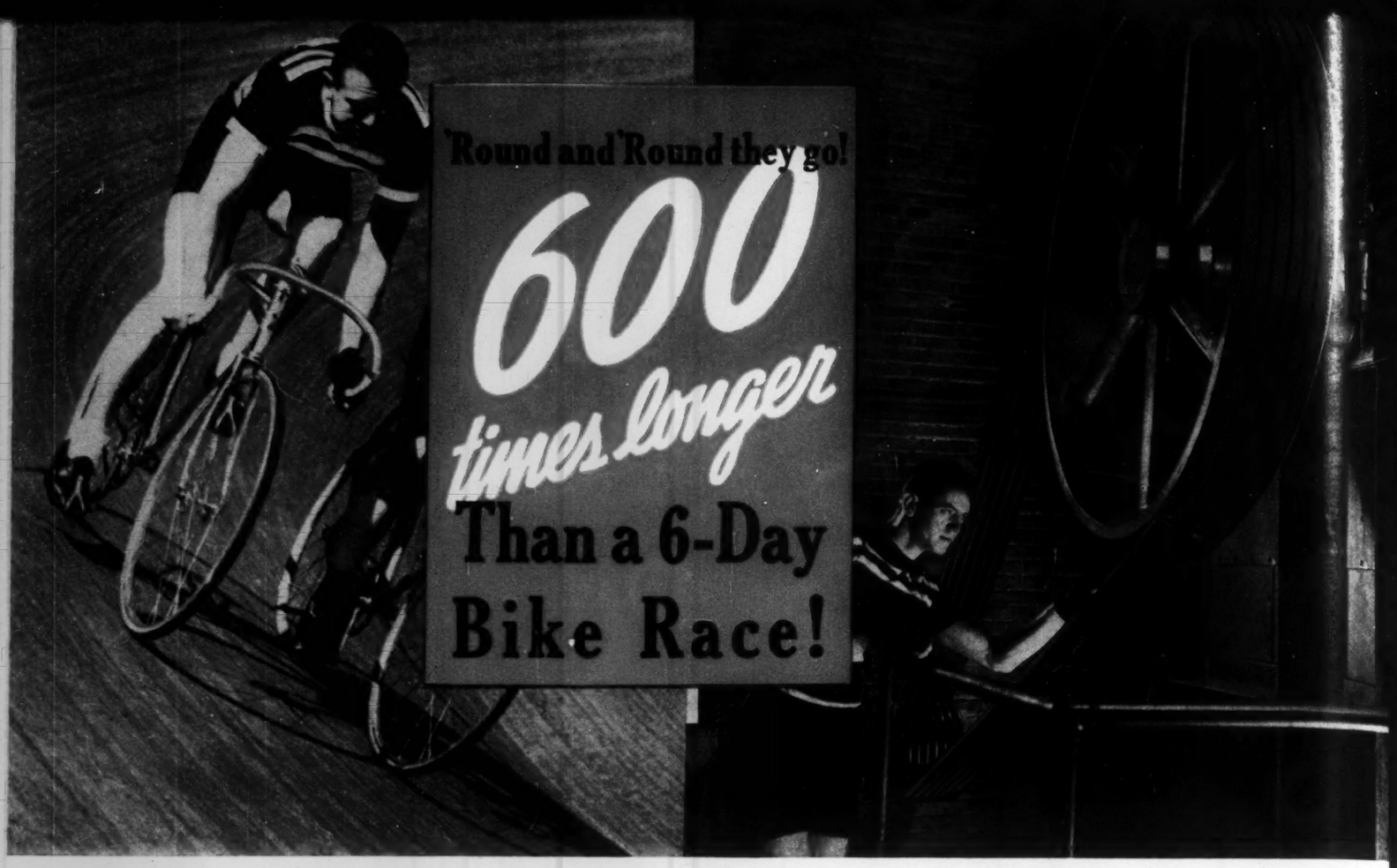

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The Editor's Pages

DEFENSE PLANTS AIR CONDITION

Increasing interest by defense industries in centrifugal refrigeration water cooling systems as applied to industrial air conditioning of airplane factories and aviation engine plants is reported by Charles E. Wilson, vice-president in charge of Worthington's Carbondale Div. Prominent among orders for spring delivery are four centrifugal refrigeration water cooling systems for the air conditioning of the new Dallas, Texas, plant of North American Aviation. The combined capacity of these four systems will provide cooling equivalent to the melting of over 5,000,000 lb of ice per 24 hr. This will provide uniformly comfortable working conditions for the men engaged in this precision work regardless of outside weather conditions. It also assures accurate fitting and functioning of all precision parts in assembly through elimination of expansion and contraction of metal due to temperature change, and by freedom from dust and tarnishing of polished surfaces.

Another of the country's largest aviation engine manufacturing plants was originally equipped with an air conditioning system when built in 1939. The results obtained were so excellent for both men and product that when the new windowless plant extension is erected this spring, three of the centrifugal refrigeration systems will be installed for the air conditioning. Each will be powered by a 600 hp synchronous motor. The three systems will have a total capacity of 1950 tons of refrigeration.

Packard Motor Corp. has ordered two of the centrifugal water cooling systems with a combined capacity of 1000 tons refrigeration for the air conditioning of their new plant where the Rolls Royce "Merlin" engines will be made for British and American fighter and bomber planes, according to Mr. Wilson.

YORK ICE PLANS NAME CHANGE

The York Ice Machinery Corp. issued a call last month for a special meeting of its stockholders to vote on a proposal which, if adopted, will effect a recapitalization and change its name to York Corp. The plan contemplates that York Ice will merge with its subsidiary, York Corp. The company stated that among the purposes of the proposal, in addition to changing the name, are the elimination of the preferred stock with its large arrearages and the facilitation of the refinancing of its funded debt which it proposes vigorously to attempt promptly upon the completion of this plan.

Stockholders of record on February 10 are entitled to vote at the meeting called for March 25.

UNIT HEATER MAKERS ELECT

The Industrial Unit Heater Association has elected A. B. Donkersley, president; C. C. Cheyney, vice-president and L. O. Monroe, secretary-treasurer for the ensuing year.

The association has been actively engaged in research and study for the development of standard methods of testing hot water unit heaters. Further study is now underway toward the extension of the standardization program, a part of which became effective June 1, 1938.

The next regular meeting of the association is to be held at Detroit on April 17.

AWARDS FOR DEFENSE SUGGESTIONS

C. Donald Dallas, president of Revere Copper and Brass, Inc., has replied to William S. Knudsen's recent plea to "forget everything except the welfare of our country" with the announcement of a $10,000 award for the best suggestions to aid in speeding up the American industrial defense program, submitted by workmen in the metal industry in this country.

"Our immediate object is to tap the vast resources of skill and experience of American workmen, which have been largely overlooked up to this time as far as any planned encouragement is concerned," said Mr. Dallas. "Specifically, what we propose to do is to offer a series of cash awards for the best suggestions to aid in speeding up America's industrial defense program submitted by foremen, sub-foremen, and men at the bench in all branches of the American metal industry."

ROOM COOLER DIVISION FORMED

A room cooler division has been formed by the Air Conditioning and Refrigerating Machinery Association, Inc., Southern Bldg., Washington, D. C., reports S. E. Lauer, ACRMA president. Membership in the room cooler division will be limited to those association members engaged in the manufacture or sale of self-contained room coolers and room air conditioners in capacities up to 15,000 Btu per hr. Principal activities of the division will be the compilation of industry statistics of sales and inventories, equipment standardization, and standard rating and testing and certification of capacities of various models of room coolers.

The announcement has a point of special interest to the air conditioning industry in stating the size or capacity of room coolers in Btu per hr rather than in horsepower. On this, the ACRMA comments: "As motor nameplate ratings, notably in the smaller sizes, are far from reliable guides, it is logical that the air conditioning industry should seek a less elastic yardstick by which its customers may measure value."

It will be interesting to observe how soon and how generally the air conditioning industry and its customers will accept this change in terminology. Habits are hard to break.

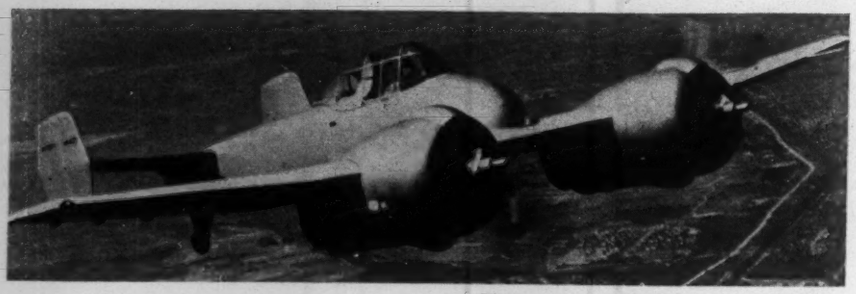

Photo courtesy Wright Aeronautical Corp.

NATIONAL DEFENSE—In our coverage of the national defense program, it is the aim of the editors of HEATING, PIPING AND AIR CONDITIONING to present information of practical value to our readers—information which is needed in the design, installation, operation and maintenance of heating, piping and air conditioning systems. An example is our lead-off article this month, which gives new data for the design of ventilation systems for electroplating tanks. Several of the electroplating processes comprise important procedures in the production of aircraft, airplane engines and other equipment vital to our defense needs. Many articles of this same type—articles with specific, usable information on heating, piping and air conditioning problems in defense work—appeared last year in HPAC and many more are scheduled for publication in 1941

QUALIFICATION FOR AN EXPERT

In a post office department memorandum pertaining to an investigation of a group offering study courses in refrigeration and air conditioning, it is brought out that the air conditioning experience of the "chief instructor" consisted entirely in having once observed as a bystander from the sidewalks the handling of certain air conditioning units intended for installation in a store. The time consumed in so doing amounted to approximately two hours.

PAGE RE-ELECTED NACA PRESIDENT

The National Air Conditioning Association re-elected Jesse W. Page, Jr., of Page-Williamson, Inc., president for 1941 at its annual convention held in Chicago last month. Other officers also re-elected are J. N. Sprekelmeyer of General Engineering Corp., vice-president, and A. C. Buensod, Buensod-Stacey Air Conditioning, Inc., treasurer. New directors of the association elected for three year terms are J. W. Bostwick of Fairbanks, Morse & Co.; H. C. LeVine of Atmospheric Control Corp.; and Mr. Sprekelmeyer.

An amendment to NACA's constitution admits consulting engineers engaged in design of air conditioning systems to full membership. A second amendment provides that associate members may be elected to the board of directors and have a vote on the board. This permits such associate members as manufacturers, for instance, who do not have a vote as associate members, to be represented on the board.

Among the new consulting engineer members are E. P. Heckel & Associates; Jaros, Baum & Bolles; Clyde R. Place; and Henry Meyer, Jr., it was announced at the meeting.

AIR CONDITIONING FOR WASHINGTON HOTEL

Although plans for the project are still in the development stage, present indications are that the total capacity of the air conditioning plant for the proposed new Hotel Statler in Washington, D. C., will be somewhere between 800 and 1000 tons, according to Holabird & Root, the architects.

According to present thinking, the refrigeration will be provided from a central chilled water plant and some type of window unit will be used in the typical guest rooms.

ANALYZE PRODUCTION, ENGINEERS TOLD

Engineers can be most useful in the national emergency by carefully analyzing production methods in their plants so that increased production and improved quality might be achieved, D. A. Newton, industrial engineer of the Carrier Corp., told members of the National Association of Power Engineers at their New Jersey state convention last month.

"Industry is now on an emergency production schedule," he declared. "Factories are being expanded, new plants are being built to produce at a greater rate than has been heretofore required. Before plant expansion is definitely agreed upon as being the one and only means of increased production, we must study the product. Under what conditions has production been better? Would this condition, the year 'round, increase production and how much?

"Air conditioning takes a decidedly important part in this step-up of production," he said. "It might be surprising to know that in some instances average yearly production has been stepped up 50 per cent, not by expansion, but by producing the product under most favorable atmospheric conditions. In addition, the quality has been improved and employee efficiency has been increased."

RESTAURANTS LEAD IN AIR CONDITIONING

Sales of both central plant air conditioning systems and electric room coolers reached new all-time high levels last year in Chicago, figures released last month by Commonwealth Edison Co. show.

For the sixth consecutive year, the highest number of central plant air conditioning installations was in restaurants. General offices and buildings ranked second on the list and drug stores third.

Largest of the 1940 central plant installations—a 590 hp job—provides controlled indoor weather for additional sections of the Merchandise Mart. Approximately one-fourth of this huge structure, or nearly 600,000 sq ft of space, is now air conditioned. Other large contracts made last year were for the Wrigley building, 400 hp; Wesley Memorial hospital, 322 hp; and Straus building, 210 hp.

NEW RESPONSIBILITIES

R. J. Resch of McQuay, Inc., air conditioning manufacturers, announced last month that Edward M. Fox has been appointed advertising and publicity manager. He will be concerned with developing new markets for several products to be announced early this year.

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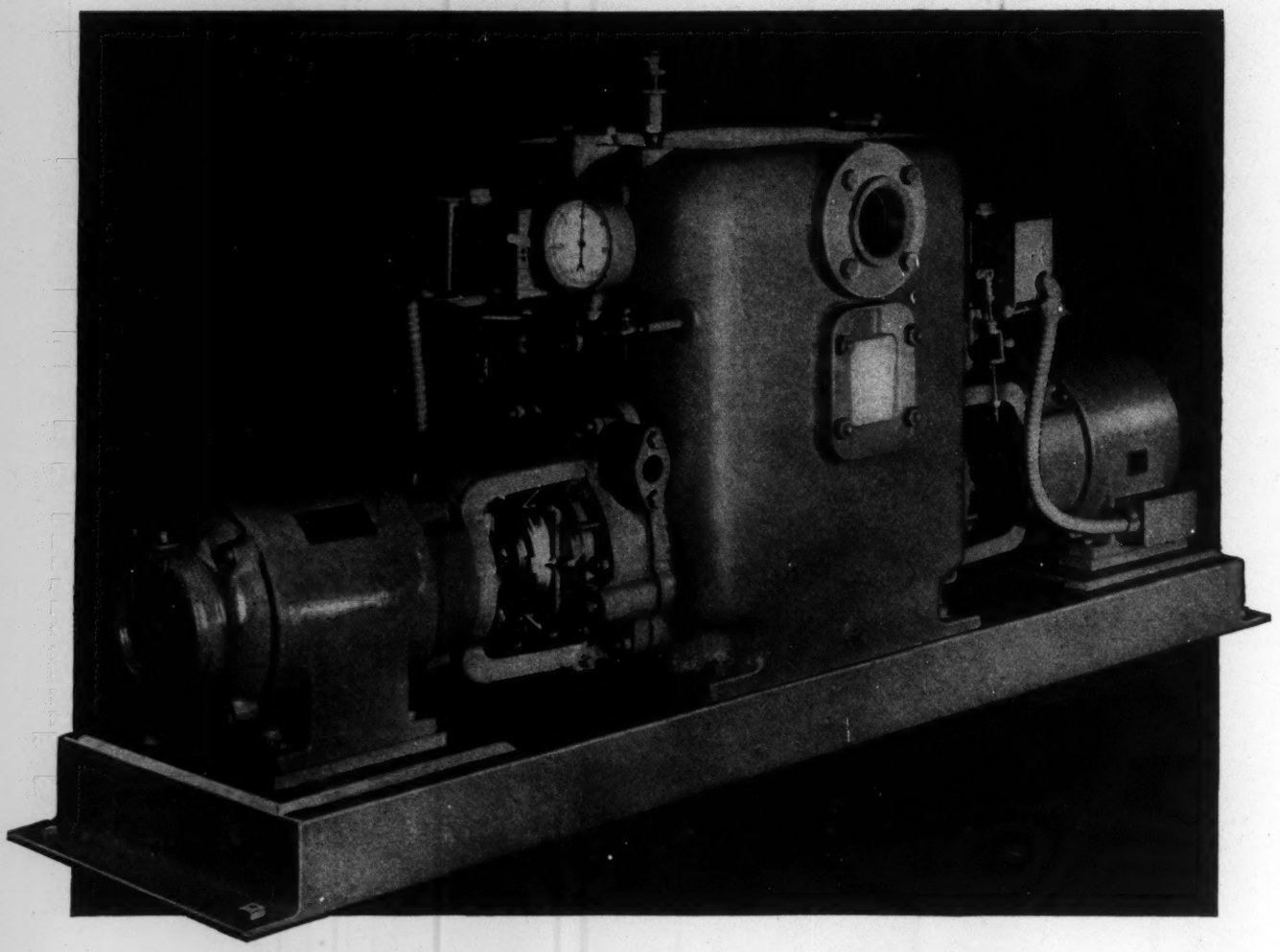

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Vol. 13, No. 2 • February, 1941

Production for National Defense

New Data for Practical Design of VENTILATION FOR ELECTROPLATING

By William P. Battista, Theodore Hatch and Leonard Greenburg*

CHROMIUM plating creates a serious health hazard through the exposure of operators to the chromic acid mist dispersed from the electrolytic solution. Control of this hazard is commonly accomplished by means of lateral exhaust ventilation through slots extending along the two long sides of the plating tank. Suitable manifolds and ducts connect the two slots to the exhaust fan and discharge pipe. The rate of ventilation recommended by several investigators [see References 1 and 2] to reduce the concentration of

*Division of Industrial Hygiene, New York State Department of Labor.

VALUABLE DATA FOR NATIONAL DEFENSE PRODUCTION GIVEN HERE

SUMMARY—It is essential in connection with some of the electrolytic processes now used in many industries to provide adequate means of protection for the health of the worker. The danger lies chiefly in the exposure to irritating and toxic mists, gases and vapors dispersed from the electrolytic solutions. The design of ventilating systems which will most effectively control these atmospheric contaminants is a practical problem for factory management, one which assumes added significance with the ever-increasing use of electroplating and anodyzing techniques in the manufacture of precision parts and machinery in modern industry. Several of these electrochemical deposition processes comprise important present-day procedures in the production of aircraft, airplane engines and other equipment vital to our defense needs.

Factors to be considered in the design of efficient ventilation for plating and anodyzing processes include current density, plating voltage, solution temperature and tank size, as well as the chemical nature of the solutions employed. Because of the fact that practically all electrolytic tanks are proportioned on conventional shapes and dimensions, certain basic fundamentals of design are applicable to all the processes wherein the dissemination of contaminants makes control by ventilation essential.

This article concerns itself with certain important fundamentals of tank ventilation with reference to proportion and design. The approach employed utilizes the usual type of ventilation practice for the control of chromic acid mists as a basis for discussion. Features of this practice have been selected as a basis for study since the toxicity of chromic acid makes it essential that these mists be controlled with a high degree of completeness, constituting in this respect a severe test of ventilation control.

Briefly, the information given here may be summarized as follows:

1) In usual practice, objectionable atmospheric contamination is removed from an electroplating tank by means of lateral exhaust ventilation through slotted air chambers or ducts along the two long sides of the tank. The capacity of the exhaust system is calculated on the assumption that the required rate of ventilation varies directly with the tank width.

2) Accepting the principle that a given minimum air velocity is required at the center of the electrolytic surface for effective control on tanks of all widths, the results of tests indicate that the rate of ventilation should vary approximately with the 1.6 power of the tank width, rather than directly with the width.

3) The rate of ventilation through the exhaust slots should be constant throughout the length of the tank. This is seldom obtained in practice, however, because of the design and size of the slotted chambers commonly used and the way in which they are installed in relation to the tank and fan.

4) Because of limited space for the installation of the ventilating ducts, it is generally not possible to employ an ideal design. In contrast to present practice however, greater uniformity of ventilation throughout the length of the tank can be obtained by the use of manifolds of uniform cross-section together with distributing vanes as specified herein.

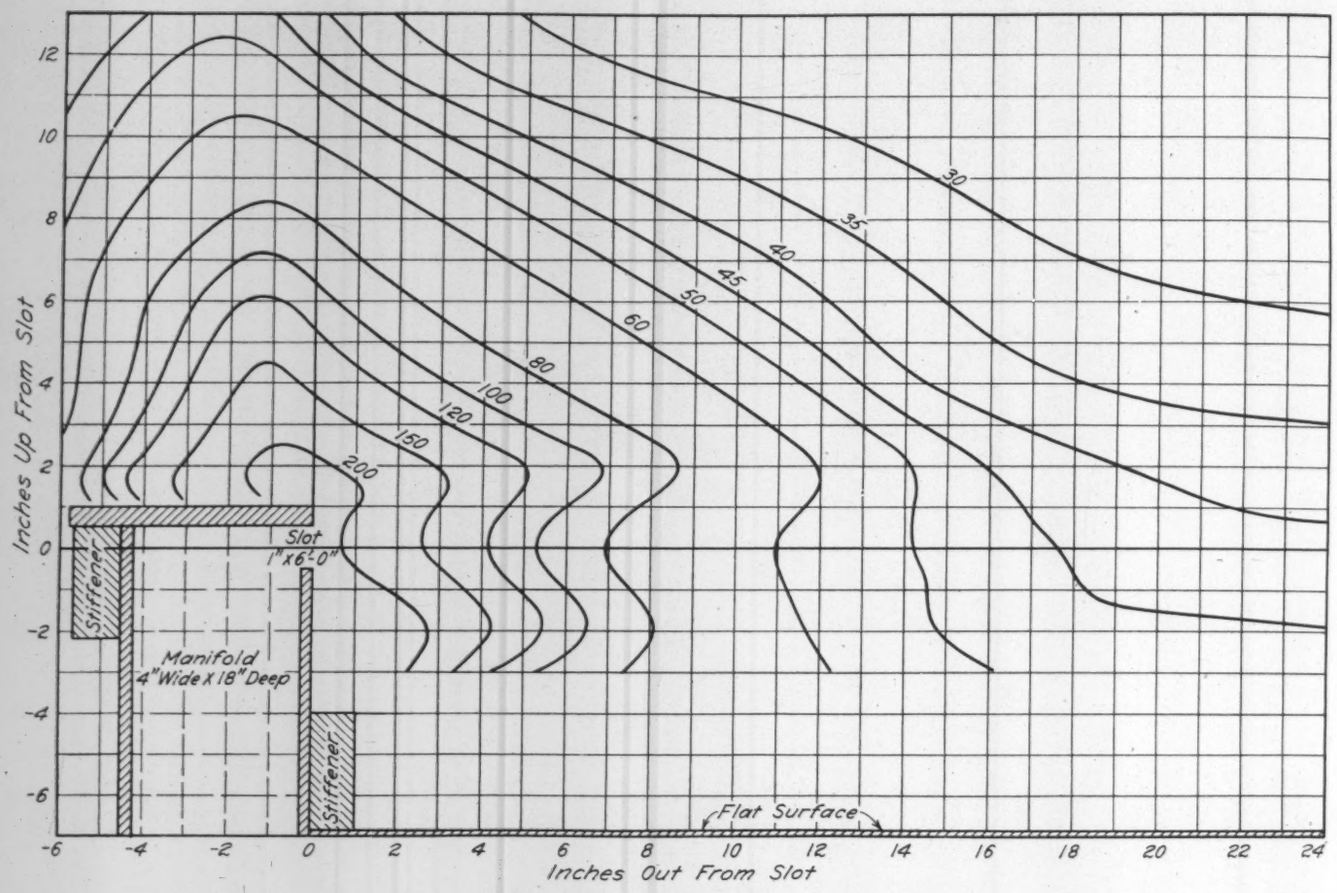


Fig. 1—Velocity contours for plating tank exhaust slot

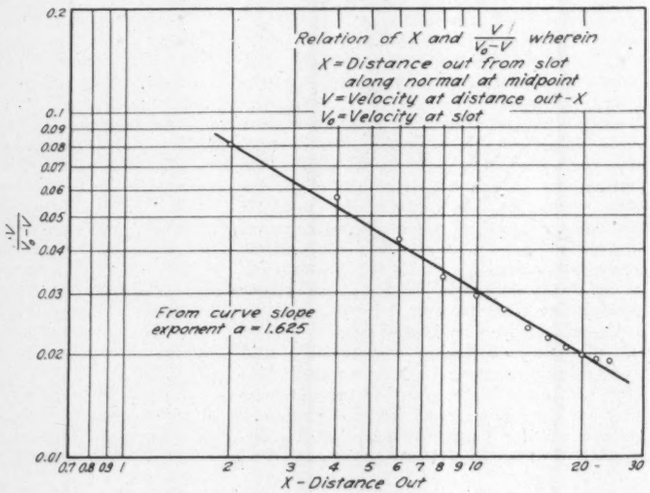
chromic acid to the safe limit is determined by the relation $Q = 100\ LW$, where Q = total rate of ventilation in cfm, L = length of tank in ft, and W = width of tank in ft.

It is clear that the rate of ventilation must vary with tank width in such a way that a required minimum air velocity will be created at the center of the tank in all cases. According to the above equation, the relationship is taken to be a direct one; that is to say, the air velocity outward along the center line of the slot is assumed to vary inversely with the distance from the slot. DallaValle's studies [Reference 3] of the velocity distribution around exhaust openings, however, have shown that the center line velocity in front of a suction opening is inversely proportional to an exponential value of the distance from the slot. This exponent was found to have a value greater than unity. In view of this, some question may be raised with respect to the correctness of the above equation for tanks of different widths. The relationship has been investigated by the authors and the velocity contours and center line velocity equation obtained for a typical exhaust system on plating tanks. The results of this study are reported here.

In addition to the necessity for a proper relationship between rate of ventilation and tank width, it is clear that the air flow per unit of tank length should be reasonably constant throughout the entire length. This follows from the most elementary consideration of the ventilation requirements. It is not an easy matter, however, to obtain a uniform distribution of flow because space limitations often do not permit the use of the elaborate exhaust piping which is required. In practice, there is need for a simple and compact ventilating system which, with reasonable power cost, will provide a constant rate of ventilation for each foot of tank length. A second objective of the present study therefore was to develop the design requirements for such a system.

Two plating tanks of different widths may be said to have equally

Fig. 2 — Relation between center line velocity and distance outward from slot


Relation of X and V / Vo-V wherein
X = Distance out from slot along normal at midpoint
V = Velocity at distance out-X
Vo = Velocity at slot
From curve slope exponent a = 1.625
X - Distance Out

effective ventilation when the velocity contours over the centers of the two tanks have the same values. Hence, the rate of ventilation must increase with tank width in accordance with the relationship which governs the change in velocity with distance outward from the slot.

Ventilation Rate and Tank Width

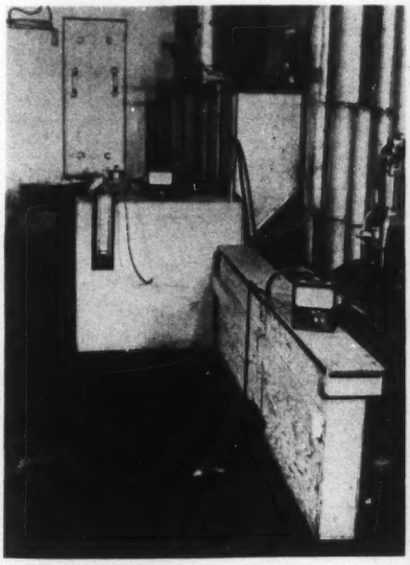
Theoretically, the velocity distribution may be considered from the standpoint of two limiting conditions: (1) a slot of infinite length or, in effect, a line source of suction of constant value throughout its length; and (2) a suction opening of such limited dimensions that the conditions approach those of a point source of suction. In the first case wherein an unobstructed line of equal suction throughout its length is assumed, the velocity contour surfaces will take the form of a series of concentric cylinders with the line source as an axis. Under this condition, the air velocity in the space around the line source will vary directly with the distance away from the line since the surface area of cylinders of equal length varies directly with their radii. Applying this to the ventilation of a plating tank, the following relation could be written:

$$Q = KLW \text{ or } Q/L = KW$$

where Q = air volume per unit time;
K = a constant;
L = tank length;
W = tank width.

This equation has the same form as the one now in use.

Fig. 3—The apparatus for studying ventilation of electroplating tanks



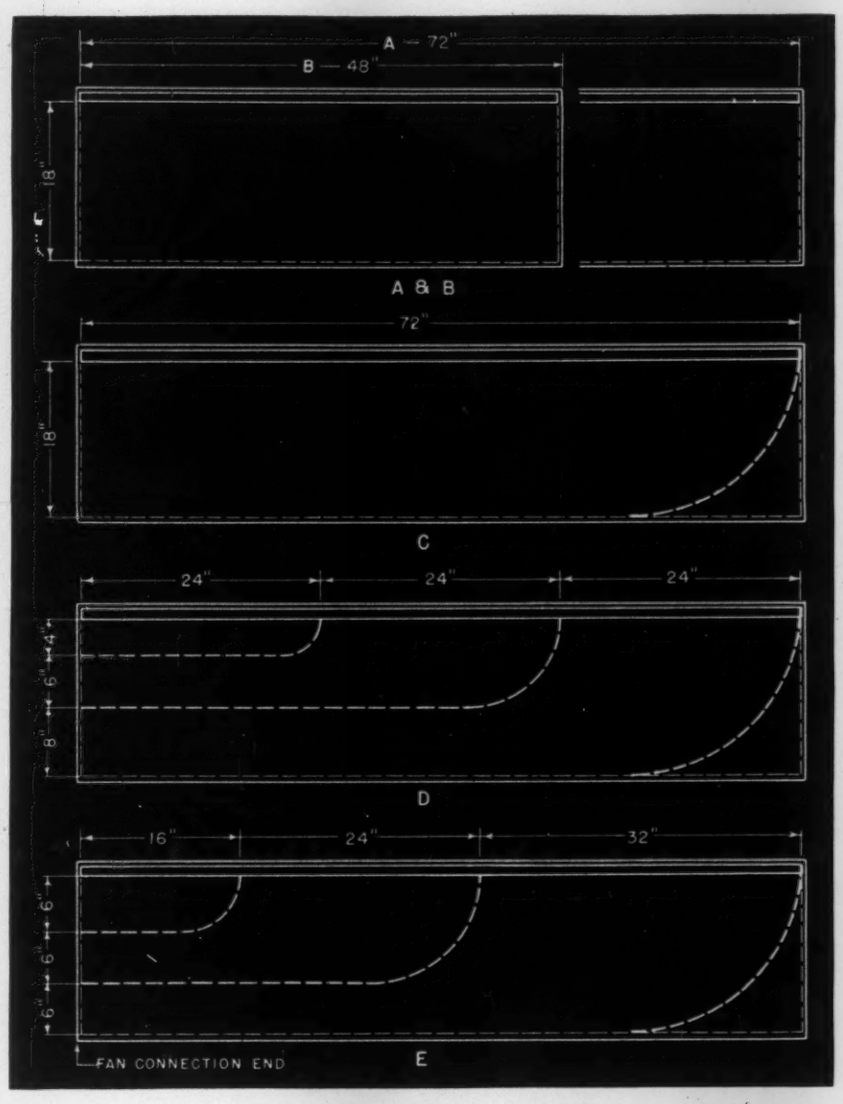


Fig. 4—Side views of test manifolds

In the second limiting case wherein approximately a point source of suction in an unobstructed space is assumed, the velocity contour surfaces will take the form of a series of concentric spheres. The surface areas of these contours will vary with the square of their radii, thus:

$$A = KR^2$$

where A = surface area of spherical contour
R = radius of contour spheres

Considering R in terms of tank width, the relation between rate of ventilation and tank width becomes

$$Q/L = KW^2$$

The velocity distribution in the actual case of an exhaust slot on a plating tank does not conform to either of these limiting cases but lies somewhere between the two. It follows, therefore, that the exponent of W in the relation $Q/L = KW^a$ must have a value between 1 and 2. For tanks of a relatively great length, a will approach unity whereas for short tanks the exponent will have a value closer to 2.

DallaValle [Reference 3] has developed the following general equation for the relation between the center line velocity V at various distances X from the exhaust opening and the velocity V_o through the opening:

$$\frac{V}{V_o - V} = KX^{-a}$$

A determination of the exact value of the exponent a for a typical exhaust slot was made by the authors on apparatus described later in this report (Fig. 3 and Arrangement E in Fig. 4). Velocity measurements were taken by means of the thermal anemometer in a vertical plane normal to the midpoint of

a horizontal slot 6 ft long by 1 in. wide having an approximately uniform rate of air flow throughout its length. A smooth flat surface placed horizontally 7 in. below the slot simulated the electrolytic surface in a plating tank.

Velocity contour curves according to accepted procedure [3] corresponding to the common slot velocity of 2000 fpm for chromium plating are shown in Fig. 1. (These contours are for one slot only. For two slots on opposite sides of the tank, the combined contours would be equivalent to the vectorial sum of the contours shown in Fig. 1 and a reverse set of the same relative values established by the second slot.) Center line velocity values, outward from the slot, were obtained from these curves and plotted according to DallaValle's general equation in Fig. 2. The slope of the straight line resulting from this plot indicates a value of the exponent a of 1.625 for the 6 ft slot. For other slot lengths the exponent will have different values between the limits of 1 and 2. In practice, however, plating tanks do not vary greatly in length and the exponential value of 1.6 may therefore be employed in the equation $Q/L = KW^{1.6}$ to determine the rate of ventilation in relation to tank width for most plating tank problems. (*Note*—The velocity distribution characteristics of exhaust slots have been subjected to more elaborate dimensional analysis by Silverman, Reference 4, the results of which will provide valuable information relative to the ventilation requirements for the many different process tanks encountered in industry.)

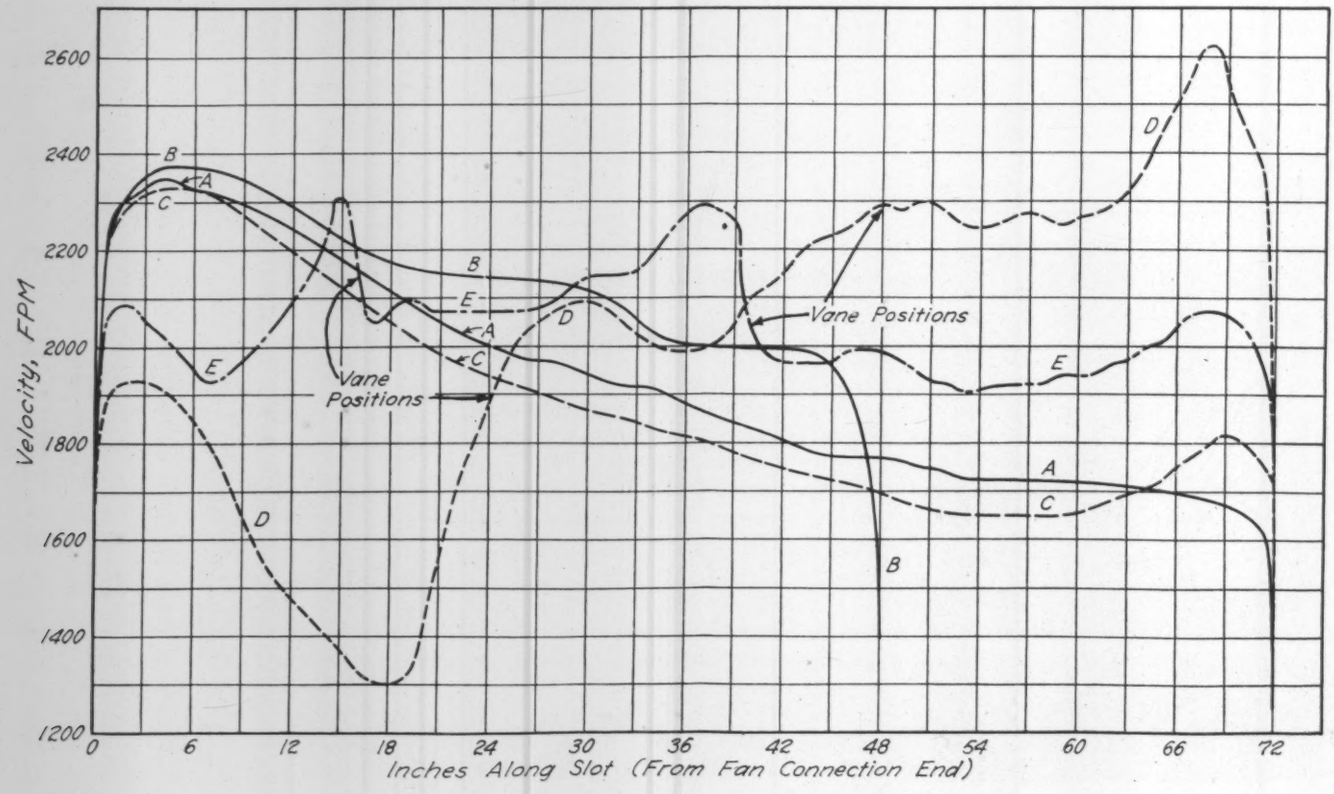
The limited data on ventilation requirements for chromium plating tanks of different widths [References 1 and 2] are not sufficient to show the correctness of the new or earlier equations. It is to be noted, however, that the rate of ventilation and the power consumption for wide tanks are greater when determined by the proposed relationship than by the equation now in use. Thus, if a rate of ventilation of 100 cfm per sq ft of tank area is necessary for a tank 3 ft wide, the exponential formula would require a rate of 160 cfm per sq ft for a tank 4 ft in width. In the interests of economy in design, therefore, further study of the ventilation requirements on plating tanks is needed, particularly with reference to the value of the constant K.

Design of Exhaust Manifold

The need for a uniform rate of ventilation through the entire length of the exhaust slot has been pointed out. This is most nearly secured by using a slotted plenum chamber of relatively great cross-sectional area in which the negative pressure is substantially uniform regardless of the location of the fan connection. Another method is to provide numerous pipe connections from the fan to equally spaced points along a slotted duct. Neither scheme is practical because of the limited space generally available around plating tanks. A tapered duct with a slot of uniform width with the fan connected to the larger end is found in practice to give a non-uniform distribution of flow. The use of an adjustable tapered slot is objectionable because the slot opening is seldom properly adjusted or fixed permanently in the correct position.

The problem becomes one of designing a compact manifold of simple construction which will provide a reasonably uniform distribution of flow and does not occupy too much space. The simplest design is one which employs a manifold of uni-

Fig. 5—Velocity distribution along slot for the various test manifolds



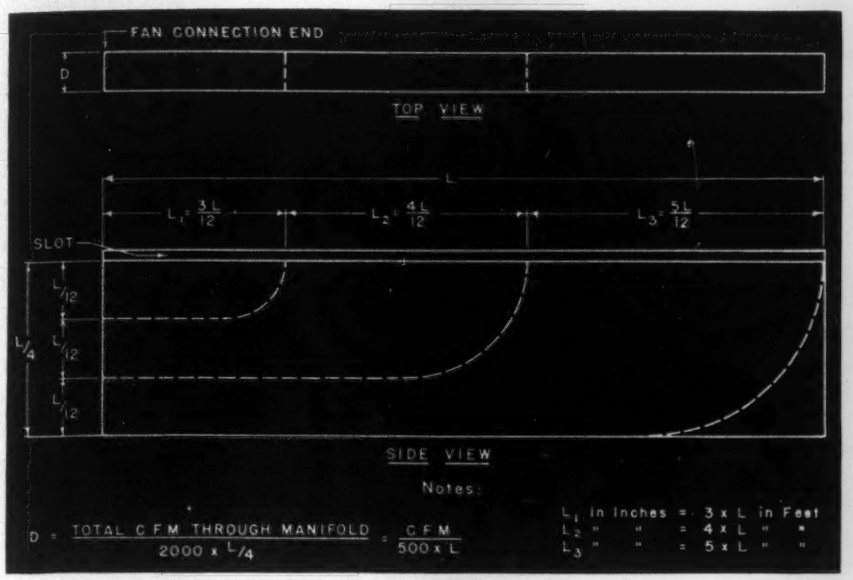


Fig. 6—Recommended design of exhaust manifolds for uniform distribution of flow

form cross-section on the two long sides of the tank with a common exhaust chamber and fan connection at one end. The cross-sectional dimensions of the manifold should be as large as possible in order to approach the effect of a plenum chamber. In accordance with common practice in ventilation, the manifold may be provided with internal vanes, as required, to insure a uniform distribution of flow. The design requirements of such a manifold are developed below.

The test apparatus, shown in Fig. 3, was fabricated of ¼ in. plywood and ½ in. lumber with stiffening members, as required. Internal dimensions of the manifold were 72 in. long, 18 in. deep and 4 in. wide. A 1 in. slot ran the full length of the manifold at the top of one side, thus providing an air intake area of ½ sq ft. A plywood box 30 in. long, 27 in. high and 12 in. wide was joined at right angles to the manifold to serve as an exhaust plenum chamber. The fan was connected to this plenum at the rear top corner diagonally remote from the manifold with common 8 in. pipe. The additional height of plenum over that of the manifold served to simulate the air flow effect of the tapered elbow ordinarily used in practice to connect the fan pipe to the two manifolds.

A blast gate on the exhaust fan was adjusted throughout the tests so that the average air velocity through the slot was 2000 fpm.

Velocities at the slot were measured with a direct reading air velocity meter using the appropriate attachment. Readings were made at the top, middle and bottom of the slot for each 1 in. interval along the slot. These were averaged to obtain one set of readings for the 1 in. slot opening.

The first series of velocity readings were obtained with the simple manifold (Fig. 4*A*). The results are plotted as Curve *A* in Fig. 5 and reveal a marked decrease in slot velocity with distance from the exhaust end of the duct. In the second test, the manifold was shortened to 4 ft, thus making its cross-sectional dimensions relatively greater and more nearly approaching the plenum chamber effect (Fig. 4*B*). The velocity distribution, plotted as Curve *B* in Fig. 5, also shows a marked non-uniformity along the slot. The results of these two tests are in agreement with past experience and indicate that a simple uniform manifold of the limited dimensions permitted in practice is not acceptable. A single curved vane at the remote end of the manifold (Fig. 4*C*) served to increase the flow in this region, as shown in Curve *C* in Fig. 5, but did not significantly improve the distribution of flow through the slot as a whole. The results of this test indicate clearly the need for additional distributing vanes.

Accordingly, two more curved partitions were installed in such a way as to divide the manifold into three separate sections, as shown in Fig. 4*D* and 4*E*. In Arrangement *D*, the vanes were equally spaced along the slot and arbitrarily located vertically so as to provide the greatest cross-sectional area in the bottom section which draws air from the remote end. The resulting velocity distribution, shown by Curve *D* in Fig. 5, indicates that too much air was drawn through the bottom section. Consequently, the vanes were relocated, as in Fig. 4*E*, to provide equal vertical spacing and a varying distribution along the slot according to simple fractional proportions. With the exception of minor disturbances in flow in close proximity to the vanes, this arrangement provides a reasonably uniform slot velocity (Curve *E*, Fig. 5). Undoubtedly, further investigation would show more desirable vane arrangements, but the practical features—and particularly the simplicity—of Arrangement *E* recommends it. The final design, with the proportions adjusted to require only the simplest design calculations, is shown in Fig. 6. The arrangement shown in Fig. 6 may be employed for manifolds of any reasonable length, provided all dimensions are determined in the proportions shown. For long tanks this requires considerable depth to the manifold. A more compact arrangement for long tanks is given by installing fan connections at both ends of the tank thus making the effective manifold length for each fan one-half the tank length.

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Organization for Defense Heating

Careful Planning Facilitates Installation of Warm Air Heating Systems at Army Cantonment

Custer is a cantonment type camp to accommodate a division

AN example of the vital need for operations management in cantonment heating is the contract for forced warm air heating in Camp Custer, Mich.

Custer is known as a cantonment type camp for the northern area to accommodate one division of approximately 19,000 officers and enlisted men and costing approximately eight million dollars. The major buildings involved are 63-man barracks (forced warm air); 45-man barracks (forced warm air); recreation or day rooms (forced warm air); officers' quarters (forced warm air); mess halls (stove heated); and storehouses (stove heated). In addition there are dozens of other buildings serving special purposes.

The forced warm air systems installed range from a unit heating a three room officer's house with a heat loss of 67,000 Btu to the double utility building which requires one 770,000 and one 760,000 Btu furnace.

Some 600 furnaces and duct systems to be installed within the few weeks allowed for completion necessitated organization which could be whipped into a smoothly functioning program in the shortest possible time and which would function without tie-ups once work began.

The vitally important preliminary problems were adequate shop facilities, a large amount of storage space, immediate delivery and continued delivery of furnaces, sheets, registers, controls, etc., a force of mechanics practically as large as could be hired, a supervisory force to keep shop and field forces operating without loss of time, and over all these, "top management" able to handle the financing, operating, and intra-subcontractor relationships so vital to the smooth completion of construction.

SUMMARY—An example of the importance and need of operations management in cantonment heating is the installation of the warm air heating systems at the army's Camp Custer, involving 63-man barracks, 45-man barracks, recreation buildings, officers' quarters, etc. Contractors' crews were organized to "streamline" the work, as described here. . . . . Success in our entire national defense effort depends upon careful planning and organization in all of its phases. This discussion shows how one important part of the defense program was carried on

Realizing that the field force must be constantly supplied on schedule, the inside work was carefully organized. Delivery assumed an important rôle, with the shop about five miles from the camp field office. Furnaces which could be spotted at the proper building on arrival were unloaded at the camp and placed outside the furnace room door. Furnaces for which there was no ready building were stored in the downtown shed. As each building was ready for the duct installation, that building duct system was loaded on a truck and delivered, with all pieces identified and the installing crew assigned for installation.

Flags Locate Crews

Green flags on the buildings were necessary to indicate the buildings in which crews were working, for even as many crews as the contractor worked could easily be lost in this large area with hundreds of buildings.

The organization of the field forces was made with the idea of dividing the work into logical operations and the crews were sized and assigned in order that all divisions of work would go ahead smoothly and according to the basic schedule.

Blower Anchor Bolts

An interesting crew was the one which drilled holes for blower anchor bolts. There was no electric current available in the camp, so the heating contractor purchased a number of portable gas engine generators to operate the portable tools. The anchor crew mounted their generator in the back of a roadster. The furnace room of the barracks has a full concrete foundation and a concrete floor—the only concrete in the building. Sheet metal templates were made so that the two man anchor crew had only to drive up to the building, lay down the template, start the generator and drill the holes.

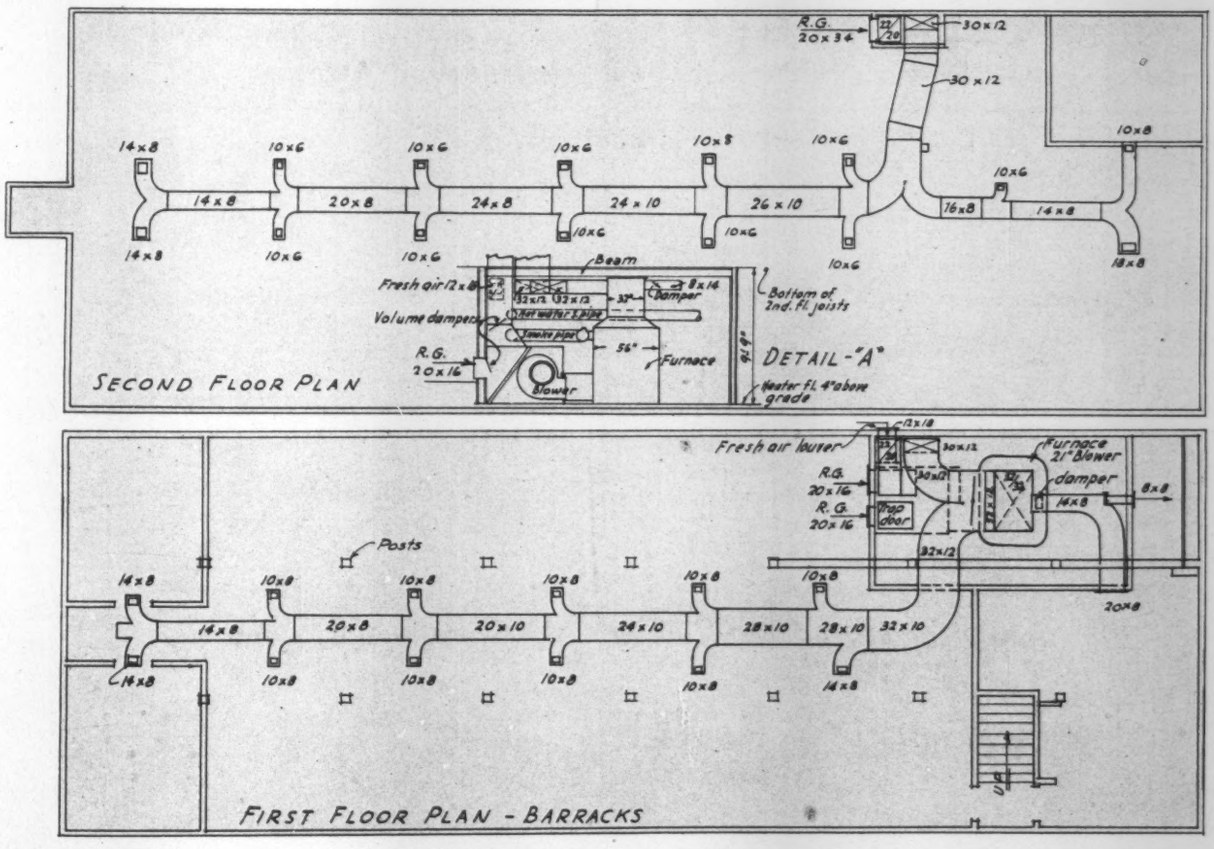
The furnace then was uncrated, set in place and the radiator hung by the labor crew.

Duct Work Crews

Two different crews of mechanics under different foremen were used to install the systems. The duct work crews (three men each) placed all duct work up to the furnace room partition. These men worked on 4 ft stepladders, their work being simplified by the nature of the duct runs and the method of dividing the duct into sections. The duct runs in barracks and small "day" buildings consist of 8 ft straight sections, one elbow, and the register fittings. Registers are taken out of the main duct in opposing pairs so this fitting was made as one section and delivered ready to hang. The registers were assembled in stubs in the shop so the duct crew had only to cleat the stub to place the register.

In the barracks, the first floor run is underneath the ceiling; on the second floor the run lies on the roof truss bottom chords. Sections throughout were put together by drive cleats on the sides and standing seam cleats on the top and bottom. The bottom was further stiffened by angles riveted to the sections. Strap hangers from the joists support the duct. Cleats were bought in 8 ft lengths and cut to

Typical system in a 63-man, two story barracks. Equipment room at first floor level. Ducts were divided into straight sections and take-off sections, thus permitting fabrication of hundreds of duplicate pieces



needed lengths in the shop and delivered in marked bundles to the job.

The duct work crew also handled one operation in the furnace room—the task of setting the breeching between furnace and outside stack. The heating contractor bought the breeching, welded in sections which could be easily handled, from a Battle Creek sheet metal contractor. Sections were put together as slip joints.

Furnace Room Crews

The furnace room crews of two or three men each (according to the size of the furnace) cased the furnace (casing furnished with furnace by manufacturer), put up the shallow canopy and assembled the plenum and duct work inside the furnace room. This crew also made the connection to the duct system and placed the canvas connection on the blower. Some small amount of cutting and fitting had to be done by the furnace room crew, chiefly because of the tight quarters behind the furnace.

The installation and fabrication of the smoke stacks was sublet by the general contractor; the subcontractor welded the 42 ft stacks into single units and erected the stacks with a traveling crane.

The installation of the controls and necessary wiring was sublet to the electrical contractor. The control system on these hand fired, coal furnaces consists of a thermostat placed in the barracks, a damper motor to operate draft and check doors, and a combined fan and high limit switch which starts the fan when bonnet air is up to temperature, stops the fan when bonnet temperature falls and starts the fan in case bonnet temperatures become dangerously high when the thermostat is satisfied.

Duct work in the furnace room was not painted, but duct work in the barracks was sprayed three coats of white by the painting contractor who sprayed the inside of the rooms.

On completion of the furnace and duct installations, the labor crew cleaned up all waste material, all unused pieces, collected all crating, paper, etc., and made the installation ready for inspection.

Coördinator

The heating contractor was Sunbeam Heating and Air Conditioning Co. In a project of such magnitude, no one contractor, of course, could operate without regard for all other contractors. In appreciation of this, the general contractor, Owen, Ames, Kimball Co., employed an engineer as coördinator. To this engineer went all claims, complaints, requests, problems and controversies. Each Wednesday night all subcontractors met with the coördinator and the general superintendent, and the past week's and coming week's work was discussed and explained. At the same time each subcontractor's progress was correlated with all other crafts so that the whole project progressed smoothly and on schedule and no one craft ran away from the others.

Types of buildings and data on the heating systems

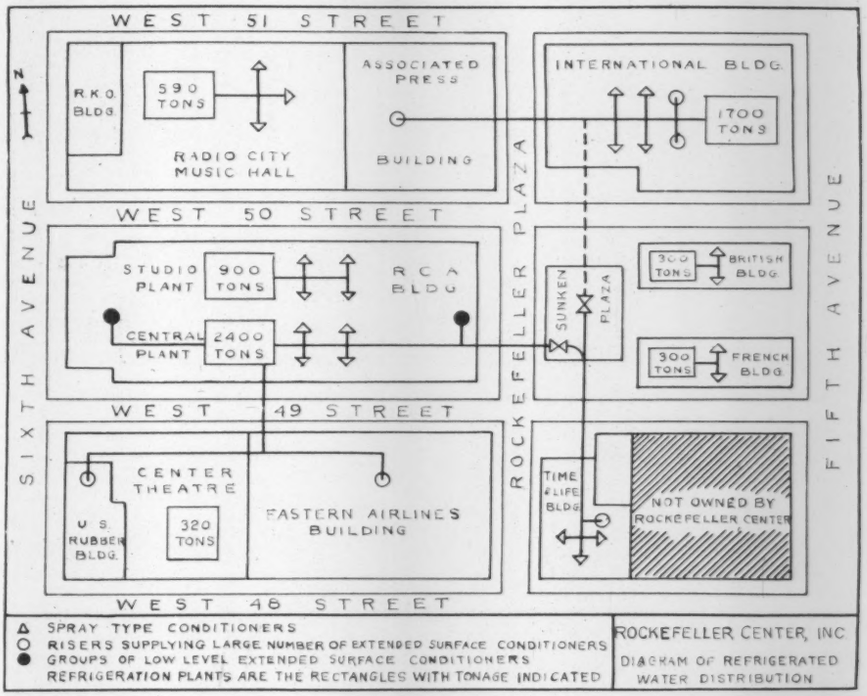
	PLAN NUMBER	TYPE of BUILDING	NUMBER of BUILDINGS					GOVERNMENT SPECS			FURNACE B.T.U. AT BONNET	BLOWER	MOTOR (SEE NOTE)		BREECHING	
			CONTRACT #1		CONTRACT #3	CONTRACT #4	TOTAL	B.T.U.	C.F.M.	S.P.		DIAMETER	H.P.	VOLTS	DIAMETER	HEIGHTS to ℄
			ORIGINAL	ADDITIONS												
	700-1171	63 MEN BARRACKS	135	20	103	57	315	446,800	5610	3/8	600,000	21"	1½	220	12"	59¼
	700-1257	OFFICERS QUARTERS-OQ-40	10		9	2	21	735,000	8050	3/8	800,000	24"	1½	220	14"	54½
	700-253	ADMINISTRATION-A-22	1				1	560,000	6810	½	600,000	21"	2	220	12"	59¼
	700-251	ADMINISTRATION-A-10	4	1	2		7	295,000	4900	3/8	458,000	21"	1	220	10"	29½
	700-299	POST OFFICE PO-2	1				1	348,000	6400	¼	458,000	21"	1	220	10"	29½
	700-330	TELEPHONE TT-2	1				1	289,000	4790	3/8	458,000	21"	1	220	10"	29½
	700-297	POST EXCHANGE E-3	4	2	2	1	9	364,000	6100	¼	458,000	21"	1	220	10"	29½
	700-310	RECREATION RB-1	5	1	2	2	10	473,000	5700	¼	600,000	21"	¾	220	12"	59¼
	700-318	UTILITY SP-11	1				1	770,000 / 760,000	4618 / 4332	½	800,000	21"	1½	220	14"	54½
	700-314	MOTOR REPAIRS SP-2	4	-2	3	6	11	505,000	8395	3/8	600,000	24"	1½	220	12"	59¼
*	700-326	STOREHOUSE SH-18	2	-1	2		3	773,000	9580	½	800,000	24"	2	220	14"	54½
*	700-378	RECREATION A-5	67		32	9	108	145,000	2400	¼	170,000	16"	⅓	110	9"	23½
	700-301	OFFICERS QUARTERS Q-7	1				1	145,000	2380	½	170,000	16"	½	220	9"	23½
	700-251	ADMINISTRATION A-7	4			5	9	220,200	3700	¼	241,000	18"	½	220	10"	46
*	700-260	GUARD HOUSE GH-2	6		2	2	10	127,000	2350	⅛	140,000	16"	¼	110	9"	23½
	700-277	FIRE STATION F-2	2		1		3	217,000	4000	⅛	241,000	18"	½	220	10"	46
	700-297	POST EXCHANGE E-2	1	1		1	3	238,000	4000	¼	279,500	18"	½	220	10"	46
	700-279	INFIRMARY I-2	5	1	2	2	10	207,000	3500	¼	241,000	18"	½	220	10"	46
	700-313	MOTOR REPAIRS SP-1	4				4	238,000	3400	⅛	279,500	18"	⅓	110	10"	46
	700-260	GUARD HOUSE GH-1		1			1	76,000	1400	⅛	97,250	12"	⅙	110	8"	24¾
	700-1179	45 MEN BARRACKS			44		44	335,000	4250	⅜	458,000	18"	¾	220	10"	29½
	700-301	OFFICERS QUARTERS Q-9			6		6	67,000	1120	¼	97,250	12"	⅙	110	8"	24¾
	700-301	OFFICERS QUARTERS Q-8			2		2	115,000	1940	⅜	119,300	16"	⅓	110	8"	24¾
	700-1257	OFFICERS QUARTERS OQM-40			1		1	735,000	8050	⅜	800,000	24"	1½	220	14"	54½
	700-1258	OFFICERS QUARTERS OQM-14			1		1	375,000	4350	⅜	458,000	18"	¾	220	10"	29½
	700-1290	GUEST HOUSE			1		1	760,000	8600	¼	800,000	24"	1½	220	14"	54½
*	700-1110	RECREATION AC				6	6	272,000	3200	¼	279,500	18"	½	220	10"	46"
			Total No. Bldgs. 11/7/40				590									

* These buildings have no water supply and no humidifier is required

Air Conditioning Keeps on Growing at the Rockefeller Center Project

C. W. Walton, Jr., Mechanical Engineer, Rockefeller Center, Inc., Outlines Increasing Demand and Use

SUMMARY—The original air conditioning arrangement at Rockefeller Center, New York City, comprised six separate plants totaling some 3000 tons of refrigerating capacity. Since that time, new buildings have been added requiring relatively more air conditioning facilities than the original buildings, and the demand continues to grow in both old and new buildings. Last season the refrigerating capacity available for air conditioning was over 6800 tons, and it appears that two or more additional 800 or 900 ton units may be needed within the next two or three years. . . . Changes in air conditioning methods during this period of expansion have therefore been brought about by a demand much larger than could have been anticipated in the initial planning eight or nine years ago, by the availability of new and improved equipment, and by the general advance in air conditioning practices. These changes and their application to the air conditioning problem at Rockefeller Center are reviewed in some detail here



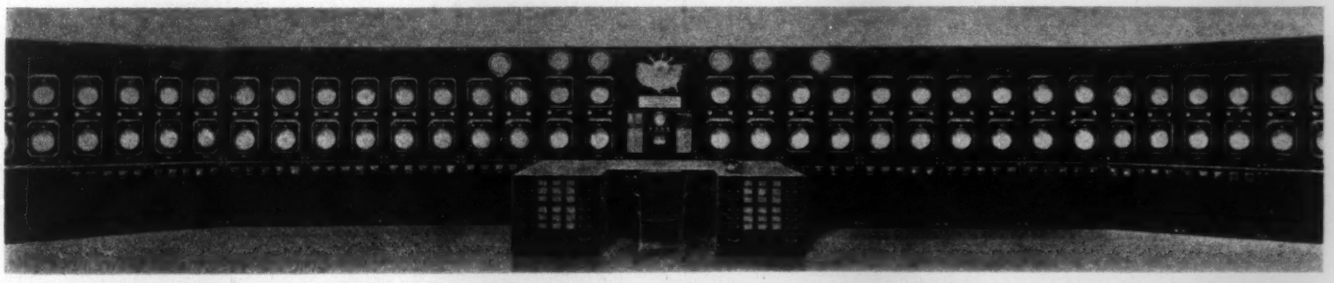
THE original air conditioning arrangement at Rockefeller Center, New York City, comprised six separate air conditioning plants totaling approximately 3000 tons of refrigerating capacity. This was nearly eight years ago. Since that time five buildings have been added, all requiring relatively more air conditioning facilities than the original buildings. The demand continues to grow in old and new buildings—with the exception of the British Empire building, La Maison Francaise, the Radio City Music Hall and the Center Theater which were fully air conditioned originally. The refrigerating capacity available for air conditioning last season was over 6800 tons, with further additions anticipated.

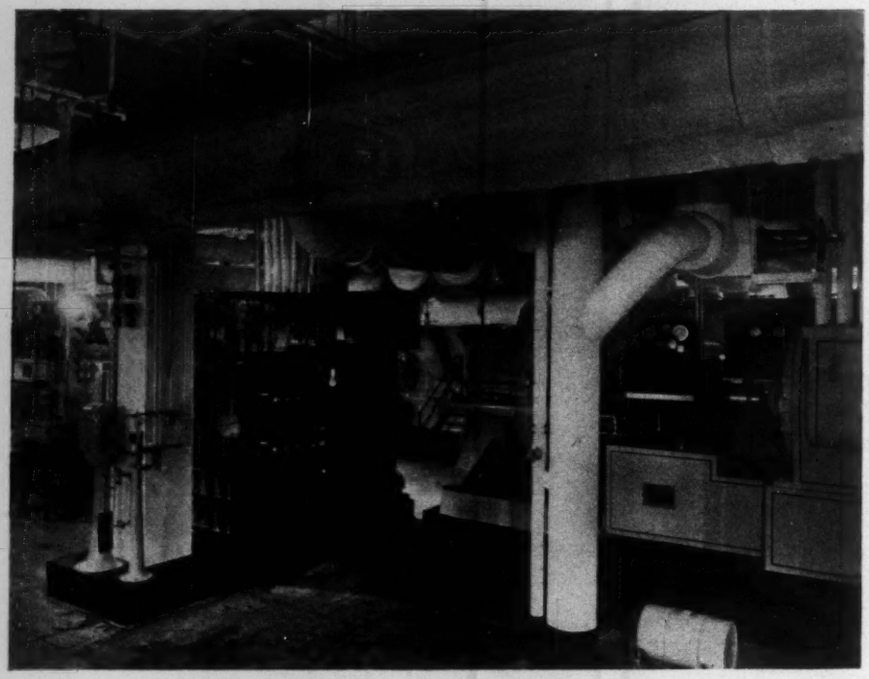
Changes in air conditioning methods during the period of expansion have been brought about by a demand much larger than could have been reasonably anticipated in the initial planning eight or nine years ago, the availability of new and improved equipment, and the general advance in air conditioning practices. These changes are discussed in detail here.

The Various Systems

Theaters—The theater systems are two of the oldest in Rockefeller Center. These are separate and complete within themselves. Each theater has three dehumidifiers and three centrifugal refrigerating units. Refrigerating units are situated in basements with the air treating apparatus above the main foyer, in each case. The Center Theater seats 3400 people and has a total refrigerating tonnage of 320. The Radio City Music Hall seats 6200 people and has a total refrigerating tonnage

These instruments and controls are needed for the NBC studio air conditioning





The original RCA building stores' refrigerating plant was enlarged as a central plant to serve some of the other buildings as well

of nearly 600. For condensing water each theater has its own group of cooling towers on its roof.

Radio Studios—The studios of the National Broadcasting Co. are served by four large dehumidifiers on the top (10th) floor of the studio section of the RCA building. Chilled water is furnished from a refrigerating plant in the sub-basement, which comprises four 225 ton centrifugal units. The dehumidifiers furnish air to numerous chambers and this conditioned outside air is mixed with return air, filtered and distributed to 27 studios. There are separate reheaters and controls for each studio system.

Stores—The stores' systems in the RCA building were served by two 300 ton centrifugal refrigerating units. This plant has recently been enlarged and will be described later. The general scheme of distribution for the stores included four dehumidifiers in the sub-basement, serving the northeast, northwest, southeast and southwest quarters; each dehumidifier then was divided into two sections with separate reheaters and fans for the main floor and concourse stores. The systems were unique in that they were completely installed for large groups of shops at the time of building construction without any knowledge of what the occupancies or load conditions would be. This refrigerating plant also supplied cold water to two large dehumidifiers for the museum of science and industry on the west side of the building and two small dehumidifiers at the extreme east for the restaurants in the sunken plaza.

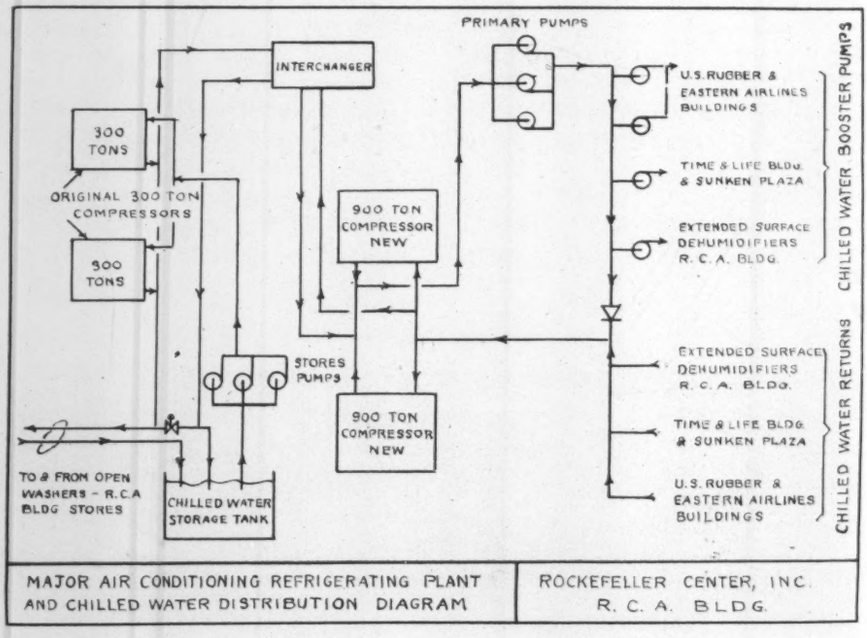
Cooling Towers

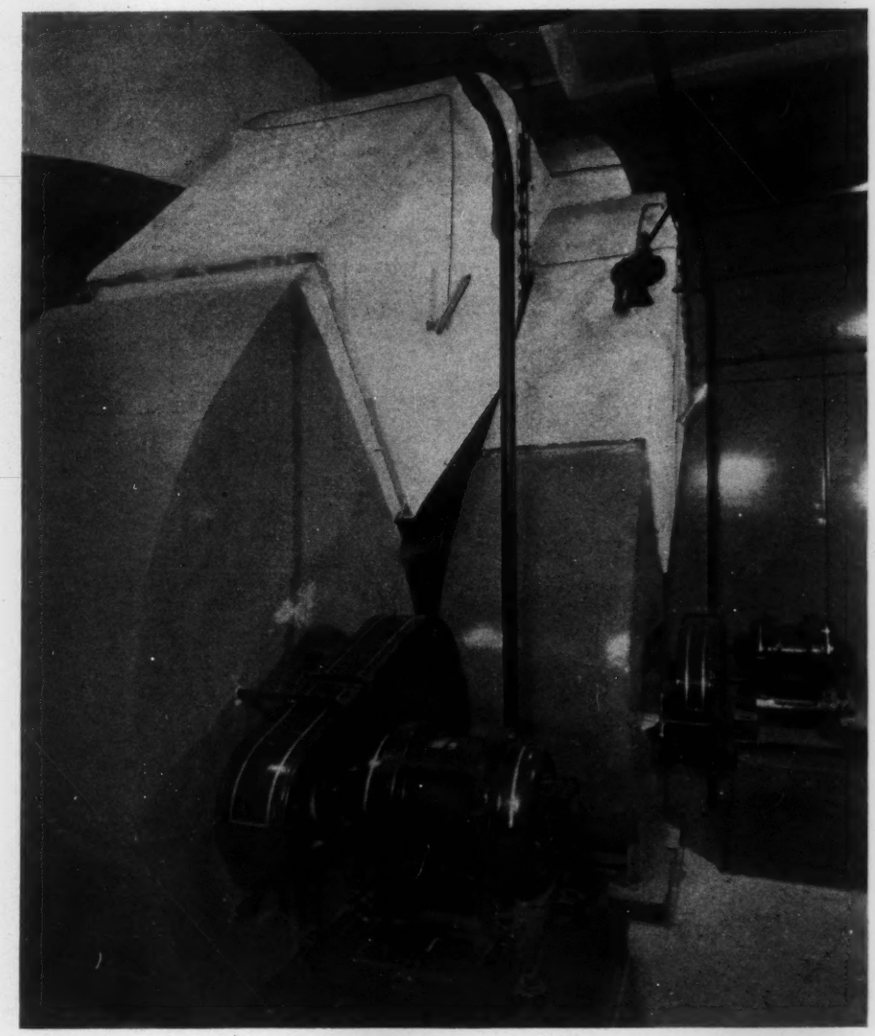
The 70 story RCA building has one lower portion facing on Sixth Ave. of 17 story height. On the roof of this are four forced draft cooling towers, two for the 900 ton NBC refrigerating plant and two for the original RCA building stores' system. There is still space available for more towers should they be required.

The British Empire building and La Maison Francaise are each equipped with 300 ton steam jet refrigerating units. Each has a stores' dehumidifier split for the main floor and concourse and an office dehumidifier split for north and south sides, second to sixth floors. On the office floors, all air is introduced through vertical supplies at each window at sill height. Returns pass through grilles, jumper ducts or transoms to the corridors. A main vertical return duct opens to the corridor at each floor. Each of the steam jet refrigerating units has its own cooling tower on the roof of its particular building. These were originally induced draft towers but were converted to forced draft after a few years of operation.

Centrifugal Machine on 58th Floor

A short time after these systems were put into operation the American Cyanamid Co. installed a system for the conditioning of their four floors in the RCA building [see HPAC, June, 1935, p. 277]; this includes a 180 ton centrifugal water vapor refrigerating unit installed on the 58th floor. This system is owned and operated by the tenant and was the first large system to be installed in Rockefeller Center which made


PRIMARY PUMPS
INTERCHANGER
300 TONS
ORIGINAL 300 TON COMPRESSORS
300 TONS
900 TON COMPRESSOR NEW
STORES PUMPS
900 TON COMPRESSOR NEW
TO & FROM OPEN WASHERS - R.C.A BLDG STORES
CHILLED WATER STORAGE TANK
U.S. RUBBER & EASTERN AIRLINES BUILDINGS
TIME & LIFE BLDG. & SUNKEN PLAZA
EXTENDED SURFACE DEHUMIDIFIERS R. C. A. BLDG.
CHILLED WATER BOOSTER PUMPS
EXTENDED SURFACE DEHUMIDIFIERS R. C. A. BLDG.
TIME & LIFE BLDG. & SUNKEN PLAZA
U.S. RUBBER & EASTERN AIRLINES BUILDINGS
CHILLED WATER RETURNS
MAJOR AIR CONDITIONING REFRIGERATING PLANT AND CHILLED WATER DISTRIBUTION DIAGRAM
ROCKEFELLER CENTER, INC. R. C. A. BLDG.



The stores' systems in the RCA building are unique in that they were completely installed for large groups of shops at the time of building construction without any knowledge of occupancies or load conditions. This shows two of the fans

use of the surface type coolers instead of open sprays.

Condenser water is cooled in a spray chamber situated in a pipe gallery between the 58th and 59th floors. Air is supplied to this spray chamber by a fan which takes air from one side of the building; the discharge from the spray chamber is on the opposite side.

"Selective Cooling"

When the International building was planned, it was decided to provide a means of supplying air cooling facilities anywhere within the building, so that tenants might be offered space either with or without a cooling system; this was in addition to the stores' system which is similar to that for the RCA building. This so-called "selective cooling" system consists of a series of supply and return risers in various columns throughout the building. A cooling system, then, may be installed at almost any point in the building; these consist essentially of cooling coils, fans, filters, small duct systems, and simple control systems. Many of the units which have been installed also have sprays for winter humidification.

These chilled water circulating systems are divided into two vertical zones so that the upper half of the building does not put an unnecessarily high pressure on the units and piping of the lower half. This system and the stores' system were served by a plant consisting of three 300 ton centrifugal refrigerating machines. Here was our first use of trichloromonofluoromethane ("Freon—11" or "Carrene—2"). The earlier centrifugal machines utilized dichloromethane ("Carrene—1"). A system of interchangers was provided to connect these refrigerating units with the selective cooling systems. Condenser water was provided by four induced draft cooling towers on the top of this 42 story building.

When the Time and Life building was built, air conditioning was planned only for the stores on the main floor and concourse levels. There was still ample refrigerating capacity available from the International building. Accordingly, three spray type dehumidifiers were installed, an interchanger was provided and chilled water lines were run through the underground trucking space from the International building plant. Shortly after the building was opened, the demand for air conditioning in some of the upper floors of the building necessitated the addition of another interchanger and supplementary piping from the International building. The cooling units installed for these upper floors are similar to those used for the selective cooling system in the International building.

Demand Expands

The planning of the Associated Press building (built 1938) indicated a more extensive use of air conditioning than in other buildings. It was considered advisable at this time that rather than build another isolated refrigerating plant, it would be better to develop the nearby International building plant to serve as a central station. Meanwhile, more and more selective cooling units had been installed in the International building so that the plant was rapidly becoming inadequate. Early in 1939 a new 800 ton centrifugal refrigerating machine was added to the International plant, making the total capacity 1700 tons [see HPAC, June, 1939, p. 369, and March, 1940, p. 162]. This was the first of the very large refrigerating units. During earlier construction 300 ton units were the largest available. At this time, a number of modifications were made in the piping arrangement to improve the flexibility for central station operating conditions. Also two forced draft cooling towers were added to

supplement the original four induced draft towers.

The air conditioning in the AP building may be considered as a combination of the ideas used in the selective cooling systems and in the stores' systems of the earlier buildings. A single pair of chilled water risers was run to the top of the building, and outside air and return air shafts were provided. Chilled water lines are paralleled by steam and return risers and compressed air piping for controls. This scheme is well adapted to moderate size installations such as the conditioning of an entire floor or half a floor, but is not as well adapted to small groups of offices as is the system in the International building. In this building, however, complete air conditioning can be offered, instead of simple cooling and recirculation. All the cooling is by extended surface coils.

More extensive use of air conditioning was indicated in the planning of the Associated Press building, which was built in 1938

The Newest Units

The newest units, the United States Rubber Co. building and the Eastern Air Lines building, are situated on the south side of the development. The provisions for air conditioning in these include chilled water risers, make-up air and exhaust air shafts, steam lines and control air lines, quite similar to the arrangement described for the AP building. More than half the office space in these two buildings is now conditioned and it is anticipated that more conditioning equipment will be added. Incidentally, there are two rather special areas in the Eastern Air Lines building which are of interest although not air conditioned; a parking garage adequate for about 800 cars and a well equipped gymnasium for the use of employees of Rockefeller Center, Inc., and affiliated companies.

Recent Enlargement

The original RCA building stores' refrigerating plant was enlarged as a central plant to take care of these southerly buildings and to take care of increasing requirements in the RCA building. The number of units added in both the International and the Associated Press buildings has so loaded the International building plant that the Time and Life building is now served from the enlarged RCA building plant. At the same time these demands came about it became necessary to provide additional air conditioning capacities in the RCA building for some new stores at the new Sixth Ave. subway station, and also for the restaurants which surround the sunken plaza. In altering and extending these systems, spray type coolers were converted to surface type coolers in order to better fit the conditions of the new refrigerating plant chilled water cycle.

To the RCA building stores' plant have been added two new 900 ton centrifugal refrigerating units each driven by a 1000 hp synchronous motor; the consulting engineer was Clyde R. Place and the contractor Buensod-Stacey Air Conditioning, Inc. The chilled water piping has been so arranged that more refrigerating capacity may be added by the simple addition of another large unit and also by the replacing of one or both of the original 300 ton compressors. Pipe sizes and general layout are such that more chilled water pumps may be added to provide for extensions of air conditioning into other areas which at present are not conditioned. A tunnel under 49th St. carries 12 in. chilled water mains to the United States Rubber Co. building and the Eastern Air Lines building, and there are two 18 in. lines to new cooling towers on the roof of the latter. The arrangement also includes an interchanger which serves as a connecting link between the new chilled water piping and that of the original 600 ton plant.

In addition to the major air conditioning equipment there are many small isolated plants, ranging from ½ ton room coolers to 20 ton plants. Some are owned and operated by tenants.

During the preparation of this description, more air conditioning units have been added, some connected to central plants, and others isolated. These amount to several hundred tons more of connected load for the coming cooling season. The installation of 12 in. chilled water mains to the 10th floor of the RCA building has been started since the accompanying diagrams were made. While it is pure conjecture on the part of the writer, it appears that two more 800 or 900 ton units may be needed in Rockefeller Center within two or three years.

POWER PIPING WELDING

Burlington Generating Station

SUMMARY—Piping welding work at the Burlington Generating Station has shown progress in the effort to improve the quality of welds, to provide the best possible working conditions for the welder and to reduce cost to the minimum, according to Philip Salmon, assistant engineer in the electric engineering department of the Public Service Electric and Gas Co., Newark, N. J. He described the details of this pipe welding project before a recent meeting of the northern New Jersey section of the American Welding Society. His talk was published in full in the Journal of the AWS, from which the information given here has been taken by permission of the AWS

THE installation at the Burlington Generating Station consists of a 100,000 kw condensing turbine-generator and two 550,000 lb per hr boilers operating with steam conditions of 1250 lb, 950 F at the turbine throttle. In welding the piping, the tendency has been to take advantage of every opportunity to simplify the work in the field in order to save time and money.

Electrodes

For instance, at Essex Generating Station a mineral coated, spiral wrapped welding rod was used for joints in carbon moly steel and a cellulose coated rod was used for carbon steel. The techniques of depositing metal from these two types of electrodes are quite different; different kinds of backing rings were used for the two processes; the scarfing of the pipe ends was not the same; and, in addition to this, it was necessary to train and qualify some of the welders for two processes instead of one. At Burlington the cellulose coated rod was used for both carbon moly and carbon steel because by that time it had been tried and found suitable for our requirements. Furthermore, the carbon moly rod was used for joints in both carbon moly and carbon steel base metal, with only one type of backing ring and one scarfing detail.

Joint Details

The joints used at Essex are shown in Figs. 1, 2 and 3. The detail shown in Fig. 3 was used for most of the joints at Burlington, except that the spacing was changed from $\frac{1}{4}$ in. to $\frac{3}{16}$ in. At Essex, it was required that all preheated welds be maintained at or above the preheat temperature from the start of welding to the completion of the stress relieving. This requirement involved costly all-night attendance on some of the larger joints which could not be completed in one working day. At Burlington the operator is permitted to stop welding and cool off the joint slowly after three layers of weld metal have been deposited. No evidence of cracks was found in the joints that were allowed to stand overnight in this condition.

Socket Joints

The socket type joint with fillet weld was used at Essex for the smaller high pressure piping, for sizes 2 in. and smaller. On the lower pressure pipe it was felt that a weld joining the relatively thin wall of the pipe to the heavier wall of the valve or fitting might burn through the pipe, and therefore a butt weld without backing ring was used. One of the causes of this trouble was the fact that most socket welding ends on valves and fittings at that time were heavy screwed end fittings bored out. By the time Burlington was started, fittings and valves with socket ends specially designed for good distribution of welding heat were available and thereafter they were used for all welded systems in sizes 2 in. and smaller.

Fig. 1—Field welds for carbon molybdenum pipe, axis less than 30 deg from horizontal, wall thickness over ½ in.

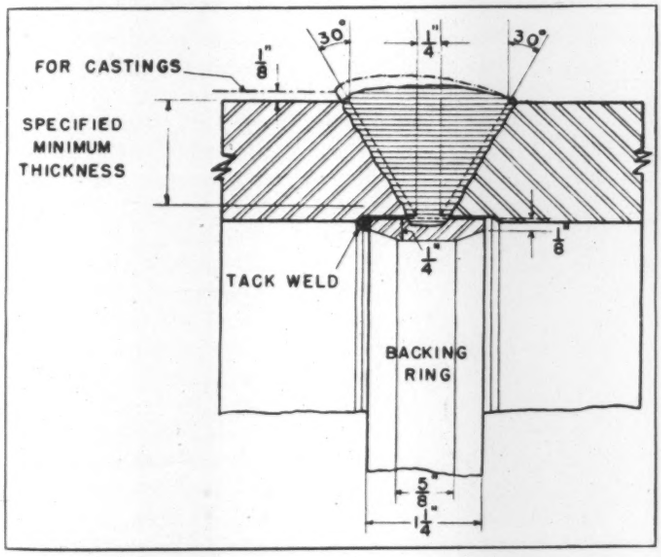
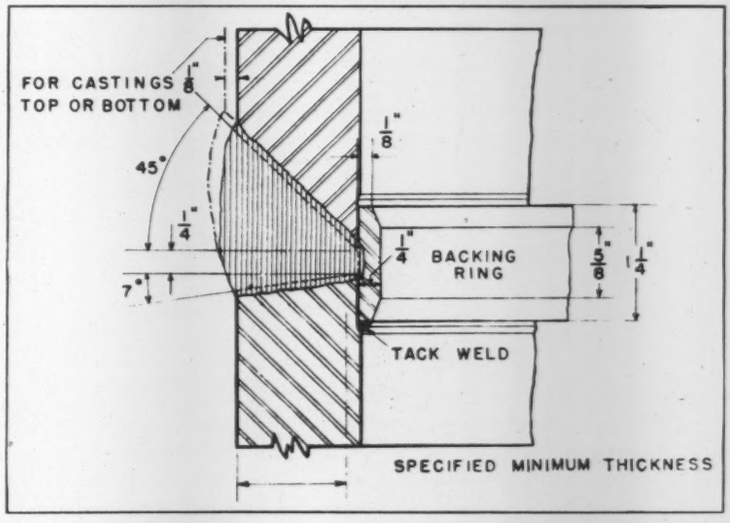


Fig. 2—Field welds for carbon molybdenum pipe, axis more than 30 deg from horizontal, wall thickness over ½ in.



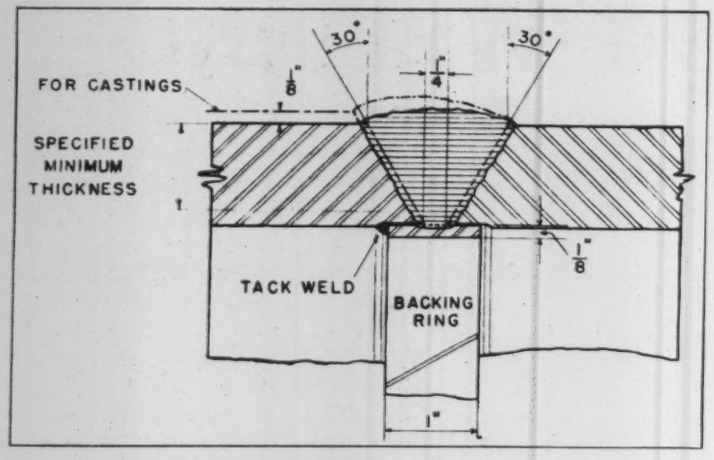


Fig. 3—Field welds for carbon steel pipe, wall thickness over ½ in.

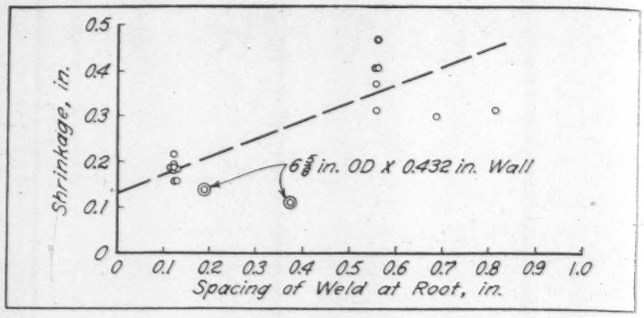


Fig. 4—Shrinkage of welds, 19¼ in. OD pipe, 1⅛ in. wall thickness

Stress Relieving

The stress relieving procedure for carbon moly at Essex was to heat the joint to a temperature between 1200 and 1250 F for one hour per inch of wall thickness. The temperature range was changed at Burlington to 1150 to 1200 F (50 deg lower) for two hours per inch of wall thickness in order to have a larger margin between the critical temperature and the stress relieving temperature.

Shrinkage of Welds

By chance an unusual opportunity was presented at Burlington to observe the effect of the width of spacing at the root of the weld on the shrinkage of butt welds in heavy wall pipe. Each of the two boilers is furnished with two downcomers 19¼ in. outside diameter with a wall thickness of 1⅛ in. These downcomers connect the upper and lower drums of the boiler with about 70 ft of straight pipe between the ends of the drums. In order to have the lower drum at the proper elevation after welding, it was necessary to estimate carefully the shrinkage in the two field welds in each downcomer. It happened that some of these downcomers were slightly short of the length necessary to line up the joints properly and consequently those joints had to be made with a wider spacing at the root of the weld. The others were furnished of the correct length and therefore we had the chance to make a comparison.

The curve in Fig. 4 shows that the shrinkage increased as the width of spacing at the root increased. The amount of shrinkage, of course, depends on various factors such as heat input, rate of welding, size of rod, etc. The two points below the curve show the shrinkage in a 6⅝ in. OD by $\frac{7}{16}$ in. wall pipe.

Welded Valves

On the Essex job, one of the six gate valves in the main steam piping was furnished with a welded joint instead of a flange at the bonnet. Since this valve has been satisfactory in service, all six of the new gate valves for the main steam piping at Burlington are of this type.

Weld Prober

The weld prober is one of the newer developments that was used at Burlington. It is a portable, power driven metal saw specially designed to cut a specimen from a weld. The device is clamped to the pipe and set for the wall thickness. One cut is made and then it is revolved through 180 deg and the other cut produces a boat-shaped piece of steel. The sample can be taken across the weld to include base metal and weld metal as shown in Fig. 6 or an all weld metal piece can be removed by starting the cut 90 deg from the position shown. The prober specimens from heavier wall pipe can be machined into a short tension test piece and tested readily. The cut is filled in with weld metal very easily. A side bend specimen was made from a rewelded prober cut in 10¾ in. OD, 1 in. wall thickness pipe with good results. The face bend test of a similar specimen taken from 12¾ in. OD, ⅜ in. wall pipe passed the requirement that no defect should exceed ⅛ in.

Position Welds

A large percentage of the field welds in a power plant job such as Burlington are position welds. The space available for the welding machines is limited and often remote from the location of the joint. Many of the welds are made from scaffolds, and it is difficult for the welder to have ready access to his machine to adjust the voltage. This condition is a hardship for the conscientious welder and it slows down the job. At Burlington a portable rheostat was provided with each welding machine so that the welder could regulate it without breaking the arc. As far as possible, pipes, fittings and valves were welded together on the floor and then hoisted into place.

The location of field welds is a matter that must be watched carefully in the drafting room. The men who make and check the drawings must be careful to avoid field welds in quarters too close for welding. Sometimes it is unavoidable.

Qualifying Welders

The qualification of welders at Burlington was much simpler than at Essex because of the changes made in the American Welding Society rules between 1937 and 1939. A guided bend tester was made on the job and the results of the qualification tests were usually known on the same day the test piece was machined. The cost of the qualification work was also proportionately reduced. The only real difficulty encountered in qualifying welders was in connection with a gas welding process which was used to join the boiler superheater tubes to the superheater outlet header. The material is a 2 per cent chromium, ½ per cent molybdenum steel. It was

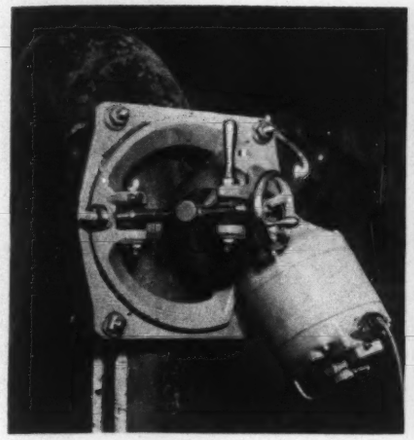


Fig. 5—The weld prober is one of the newer developments used

necessary to make some changes in the prescribed heat treatment of the weld before the specimens could be made to pass the guided bend test. These joints are in a difficult location for welding and it was very important that the welds be just right. In order to be sure that they were right, the hardness of these joints was tested by means of a portable hardness tester. The device is clamped to a joint and a hardened steel ball is impressed on a ground spot until a predetermined pressure is attained. The diameter of the impression is a measure of the hardness.

As required by the boiler code most of the field welds made on the boilers were inspected either by gamma ray or by the magnaflux method. All of the joints but two, which were so inspected, proved to be entirely satisfactory. One of these was a surface defect and the other was a slag inclusion.

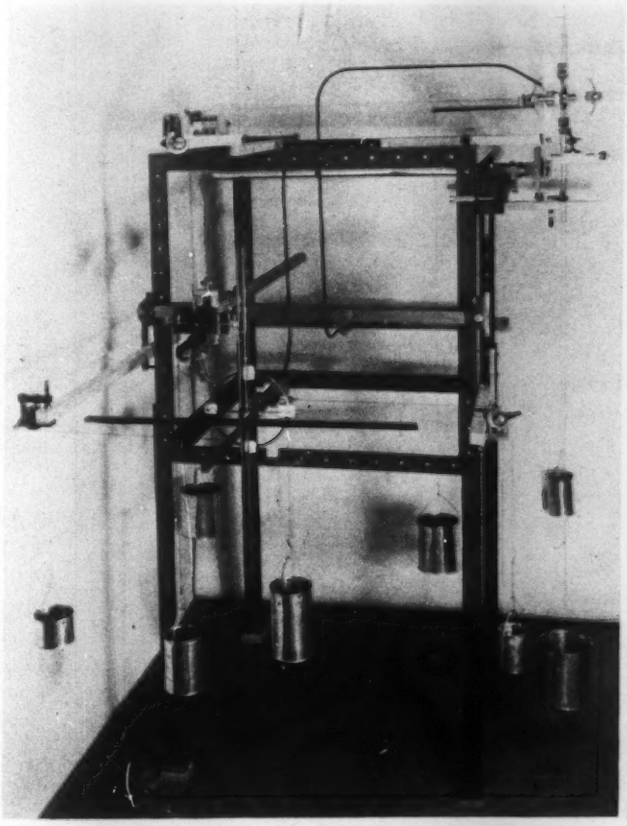
In conclusion, it may be said that the welding work at Burlington has shown progress in the effort to improve the quality of the welds, to provide the best possible working conditions for the welder and to reduce the cost of welding to a minimum. Further improvements are predicted as time goes on.

Fig. 6—Cuts made by the weld prober

Models Solve Piping Problems

FRED M. HILL, Commonwealth & Southern Corp. engineer, described a novel mechanical method for solving complex problems involving stresses and reactions in high pressure steam plant piping in a paper presented at the 61st annual meeting of the American Society of Mechanical Engineers. It enables accurate and reliable solution of complicated pipe networks without the use of laborious and complex methods of mathematical computation.

The mechanical method was developed by Mr. Hill to facilitate the solution of such problems by the use of models employing small, solid, die drawn steel rods on the order of ¼ in. in diameter, which have been found to possess advantages in being easy to handle and form to shape and in yielding readily measurable deflections with the application of small loadings.



Model set-up of a three dimensional pipe run

Another advantage is the possible exaggeration of deflections to the extent that one may easily observe the effect of various changes in, or the relative importance of, the applied loadings. In this respect the testing device may be thought of as an analyzer. Although the rods do not simulate the prototypic pipe with respect to the flattening or ovalization occurring within the bends the close agreement between model and mathematical answers appears to justify their adoption.

Consideration was given to the use of tubing with its theoretical advantage of closer simulation but the practical advantages of rods were decided to be of greater importance.

The method is also useful in cases where complexity of the problem is such that no mathematical solution has yet been produced.

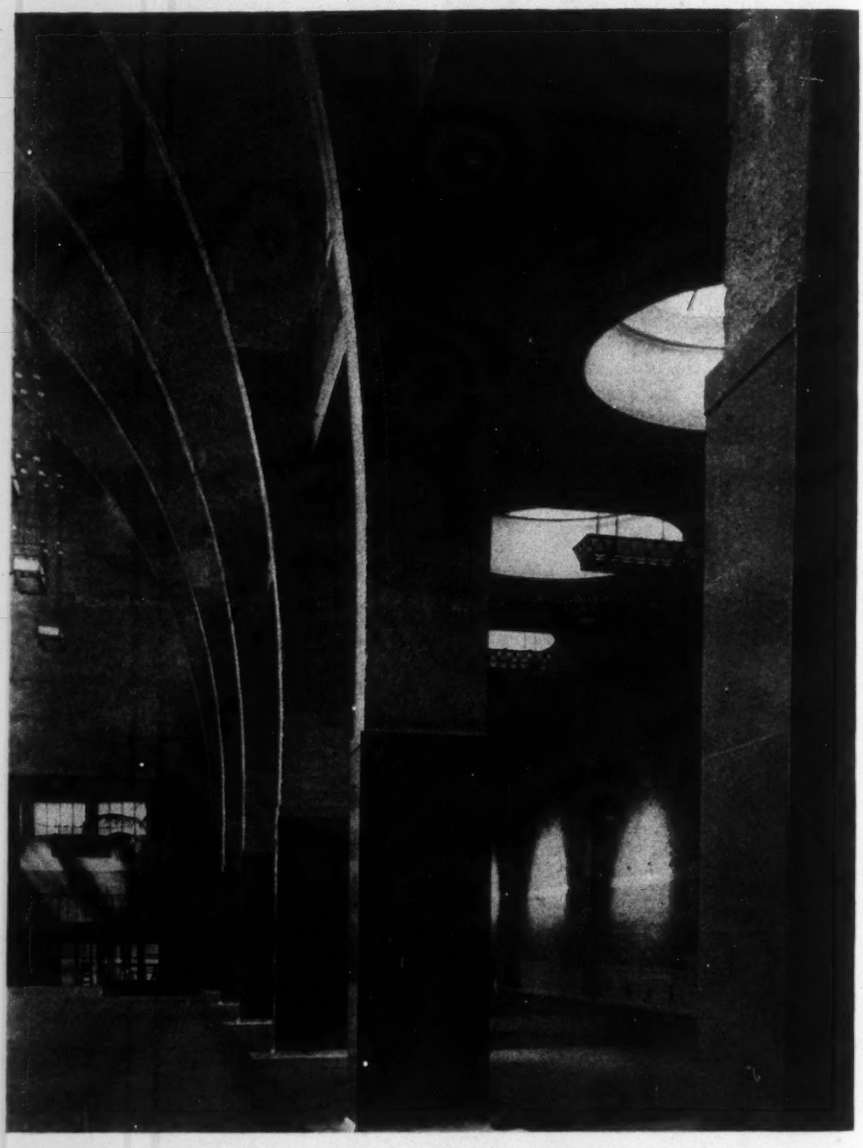
Panel Heating Throughout the World

SUMMARY—So much is heard of panel or radiant heating these days that it seems worthwhile to pause for a look at its use or acceptance throughout the world. We've therefore asked Mr. Hutchinson, instructor in the Department of Mechanical Engineering at the University of California, to analyze the installations in the U. S. and abroad by types of buildings, pointing out the reasons for its applications in the various places where it is used. Mr. Hutchinson was in Europe in 1938 and 1939, where he made a rather extensive study of the subject and of panel cooling. . . . . His article emphasizes the extent and diversity of existing installations. It is pointed out that the uniform temperature gradient with this type of heating is a principal reason for its application to high ceilinged rooms in churches and banks, that an advantage in factories is the possibility of obtaining local regions or zones of comfort in a space where the general temperature level is low, and that in hospitals and schools it provides a means of establishing comfort while still retaining an open air environment. The reasons for the wider acceptance of panel heating in large buildings as compared with residences are also presented

PANEL heating has recently gained considerable attention in the United States, but it is perhaps not yet realized how numerous and diversified are installations of this kind in Europe. The sinuous coil type of panel originated in England and a commonly held belief in the United States is that its development and application have been limited essentially to the British Isles. The list of panel installations which follows is, in itself, substantial evidence to demonstrate the fallacy of the above belief; in addition, it suggests a relative freedom of this system from political, geographic or climatic limitations:

- Headquarters of the Nazi Party, Munich.
- Benito Mussolini Tuberculosis Hospital, Rome.
- Cafe de la Paix, Paris.
- Palace of H. H. Maharajah Gaekwar of Baroda, India.
- Hong Kong and Shanghai Bank, Hong Kong.
- Oslo Town Hall, Oslo.
- City Library, Wiborg, Finland.
- Banque d'Espagne, Madrid.
- Residence, Buenos Aires.
- Bank of England, London.
- British Embassy, Washington, D. C.

Panel heating also travels the seas in three shipboard installations, one of which—on the liner "Queen Mary"—makes use of two different types of panels: the sinuous coil method and the direct electric resistance fabric method.



The 80 ft vaulted ceiling of the Royal Horticultural Hall, Westminster, is heated by means of hot water coils embedded in the concrete ceiling slabs between the arches.

Table 1, based on data supplied by the Invisible Embedded Panel Warming Association, London, shows the extent to which the sinuous coil system has gained worldwide acceptance. Vertically the tabulation permits determination of the distribution of installations among countries, while horizontally it presents a breakdown, for each country, in terms of the type of structure. It will be noted that there are a total of more than 1200 existing installations in 21 countries (Austria and Czechoslovakia are included with Germany; Ireland and Scotland with England).

To a greater extent than is true

Types of Buildings Using It and Reasons for Its Use Analyzed by F. W. Hutchinson, University of California

of other types of heating, the reasons justifying selection of a radiant system vary widely with the type of structure for which the installation is proposed. Readers of HPAC are already familiar with the principal operating characteristics of low temperature radiant heating. These include a low floor to ceiling air temperature gradient, and establishment of equivalent comfort in an atmosphere having an air temperature 5 to 7 F below that which would be needed if convective heating were used.

Considering the above characteristics in connection with some of the structures of Table 1, it should be possible roughly to evaluate the relative advantage of radiant heating for buildings of different types.

Banks and Churches

The first two columns of the table cover banks and churches. Buildings erected for either of these purposes are usually characterized by high ceilings and it would therefore be expected that the choice of radiant heating for such structures would be based essentially on the desirability of reducing the temperature gradient. A numerical example will demonstrate the extent to which this is true.

For a radiator heated room with a 20 ft ceiling the Heating, Ventilating, Air Conditioning Guide of the ASHVE recommends evaluation of the mean temperature in terms of a 2 per cent increase for each foot of height above the breathing level. On this basis the temperature gradient in such a room would be $0.02 \times 20 \times 72 = 28.8$ F and the mean temperature (at the 10 ft level) would equal $72 + (5 \times 0.02 \times 72)$ or 79.2 F. Infiltration air would therefore have to be heated from the outside temperature to 79.2 F and for a mean seasonal outside temperature of 50 F the rise would be 29.2 F. If this same room were panel heated, the mean inside air temperature would be approximately equal to that at the breathing level and the required infiltration air temperature rise would be 22 F. But the temperature at the breathing level with a panel system can be 5 F lower so the actual mean temperature would be 67 F and the infiltration air temperature rise 17 F. Thus the infiltration loss with panel heating would be less than that of a radiator system by $12/29 = 42$ per cent of which $7/29 = 24$ per cent is attributable to the more uniform temperature gradient. (For mean outside temperatures other than 50 F the percentages given would of course vary.)

In addition to the indicated reduction in ventilation load, there would also be an operating saving resulting from decreased transmission losses; the magnitude could easily be determined for a particular case, but is not directly calculable from the air temperatures.

The economy resulting from reducing the temperature gradient is obviously proportional to the height of the ceiling. It becomes most significant in auditoriums or in the entrance halls of public buildings. A striking example of this is the 80 ft vaulted ceiling of the Royal Horticultural Hall, Westminster, which is heated by means of hot water

Number of Panel Warming Installations (Crittall Patents) to 1938—Hot Water Sinuous Pipe Coils Only

Countries	Banks	Churches	Factories	Flats	Hospitals	Hotels	News Bldg.	Offices	Homes	Public Bldg.	Schools	Stores	Ships	Total
England	21	23	10	21	116	22	12	69	133	101	97	81	3	709
France	3	17	7	7	32	18	2	36	83	17	20	10		252
Italy	2	4	2		12			12	14	9	1			56
Germany		2		2	4			10	22	3	4	3		50
Holland	1		1		8	1		5	4	1	1	7		29
Switzerland			2	1	10			2	10	1		1		27
Sweden				7	2	1		2	2	6	3			23
Denmark	4		1	3	2	2			2		9			23
Belgium		1		1				1	8	1		2		14
Spain	1				1			1	2	4				9
Bulgaria				1					5					6
Norway					1			1		3		1		6
Algeria					3				1					4
U. S. A.					1				1	2				4
Finland				1						2				3
India									1	1	1			3
Argentina			1						2					3
Hungary									1					1
Uruguay					1									1
China	1													1
Egypt								1						1
Total	33	47	24	44	193	44	14	140	291	151	136	105	3	1,225

coils embedded in the concrete ceiling slabs between arches.

Factories

The importance of reduced temperature gradient would not be so great in the case of factories (third column of Table 1), but here too it is frequently a factor. A more important advantage of radiant heating for warehouse, factory or other large open structure installations is that it enables establishment of local regions or zones of comfort in a room for which the general temperature level is low. This characteristic of panel systems is important whenever the density of occupancy is low or non-uniform. In this respect the use of panels is somewhat analogous to the use of luminaires to increase the intensity of illumination in parts of an office for which the general level of illumination is insufficient.

Hospitals and Schools

With hospitals and schools, an important advantage is the possible reduction in ventilation heating load. High ventilation rates are commonly used in such structures and hence a marked economy would be expected from a 5 to 7 F reduction in air temperature. More important, however, is the fact that panel heating provides an effective means of establishing comfort while retaining the advantage of an open air environment. In some hospitals, panel heated wards are completely open to the atmosphere on one or more sides. Authorities at the Winford Orthopaedic Hospital in London state that their patients are comfortable in such wards even when the inside air temperature is as low as 53 F (with an outside temperature below freezing).

Stores and Offices

For stores, offices and public buildings an advantage frequently claimed is that the lower inside air temperature reduces the shock accompanying short time occupancy of any heated or cooled space. A health point sometimes raised in this connection is that the relative humidity in a panel heated structure does not drop as low as in buildings heated by other systems.

Residences

Table 1 shows that of 1225 panel heated structures 291 are residences. This fact is of particular interest. It suggests a much more rapid acceptance of the panel system for large in contrast with small structures. Since the number of residences exceeds by many times the number of buildings of all other types, it is perhaps significant that there is only one residential installation for three in larger buildings. It is not within the scope of this article to attempt an evaluation of the factors responsible for the above circumstances, but the more important of the causes which are thought to contribute are listed below:

1) Central heating is not commonly used in European and English homes. While the potential field of application to residences is much greater than that to other kinds of buildings, the actual market for heating is largely in business or public structures.

2) The relative cost of a panel installation is greater for small than for large buildings because:

a) Heat losses from the rear of the panel are greater. In multi-story buildings the entire heat input to all ceiling panels except those in the top story goes to the structure; in single story buildings the loss through the roof is obviously much greater. This disadvantage can be offset by insulation, but only at greater capital cost.

b) First and operating costs of a panel system are least when it is operated continuously. For small structures intermittent operation of the heating plant is preferable, but with panels this would necessitate the provision of considerable additional surface for start up and would introduce control problems due to the greater thermal capacity of such a system.

Summarizing, it may be said that panel heating is especially suitable for buildings with high ceilings and high rates of ventilation. As these two factors are reduced the value of lower temperature gradient diminishes until, for a low ceilinged store or residence, it is of no significance. The value of the 5 F or more reduction in mean air temperature is, of course, valid for any type of building.

Panel Heating in U. S.

One more item in connection with Table 1: The position of the United States, between Algeria and Finland, is noteworthy. Why has the acceptance of panel heating been so much slower in the U. S. than in other countries? Evaluation of the reason is difficult, but some contributing factors are listed below

1) Central heating was commonplace throughout the United States before the development of the panel system. In consequence,

a) There was not the same incentive to the development of a new system.

b) Competition with other methods would have been much greater than in Europe where central heating in any form was largely unknown.

2) People in this country were used to the equipments of convective heating and do not object, as do the English, to the presence of visible heating elements.

3) Central heating has for so long been an essentially American development that it requires some effort to recognize a change in method which originated in a country long recognized for the inadequacy of its heating methods.

4) Since the panel system is designed for continuous operation it is well adapted to the use of offpeak electrical energy. This fact has contributed to the popularity of the panel system in England since unusually low rates have been available there to users of large blocks of offpeak energy.

In conclusion, attention is again called to the fact that Table 1 covers sinuous coil embedded panel installations *to* 1938 only.

Other methods of panel heating are widely used throughout Europe. Among these are the cast iron plate type and the electric fabric type. The total number of complete panel installations is in excess of 2000 and it has been estimated that an equal number of individual detachable panels are in service.

PAYS FOR ITSELF

Soot, dirt and noise from passing railroad trains—which had proved a detriment to business—have been practically eliminated since installation of air condition in the Larkey Co. men's clothing store in Passaic, N. J. The store occupies the first two floors of a building directly across from the railroad, which runs along Passaic's main street. A 15 *per cent increase in the number of customers and a* 10 *per cent increase in the number of sales is attributed by owners of the men's clothing store to the installation of air conditioning. At the present rate, it is estimated the installation will pay for itself within three years. In addition, there has been a* 15 *per cent reduction in employees' errors, a* 10 *per cent decrease in absence due to illness and a* 10 *per cent decrease in loss of time from closing early in hot weather.—Edward T. Murphy, Carrier Corp.*

How to Figure the Dust Concentration When Filtering Process Is Continuous

By Frank B. Rowley* and Richard C. Jordan†

SUMMARY—The authors have made studies on atmospheric dust and methods of measuring it for several years, but until now nothing of a semi-popular nature about the various factors involved, or the developments which have been taking place in this field, has been published. It is the authors' feeling that the question of atmospheric dust has been more or less neglected insofar as methods of measurement or standards are concerned—yet it is most important in many air conditioning problems, particularly now that industrial activity is increasing and will continue to do so under the requirements of our national defense program. . . . . The factors affecting dust concentration levels in enclosed spaces were explained in the December HPAC; it was pointed out that the rates at which dust laden air is introduced into the space, with which dust is produced in the space and with which dust settles from the air, and the rate at which air is filtered, are usually of greater importance in the determination of actual dust concentrations than the efficiencies of the filters themselves. This month, the application of the data to continuous filtering processes is given. . . . . The theory upon which this discussion is based was proposed by Dr. Jordan

THE various factors affecting the dust concentration levels in enclosed spaces were analyzed in the December HPAC; it is now possible to derive equations involving these factors to be used for the prediction of dust concentrations in such spaces. Many different combinations of conditions are encountered in practice, and it is therefore advisable to make the equations as general as possible. The specific equations required to handle individual cases can be determined from the general equations merely by eliminating unnecessary factors.

Referring to Fig. 1, which shows a typical filtering installation and indicates the various gains and losses of dust, let—

L = infiltration air, cfm;
M = exfiltration air, cfm;
N = recirculated air, cfm;
F = outside air, cfm;
E = filtering efficiency, per cent;

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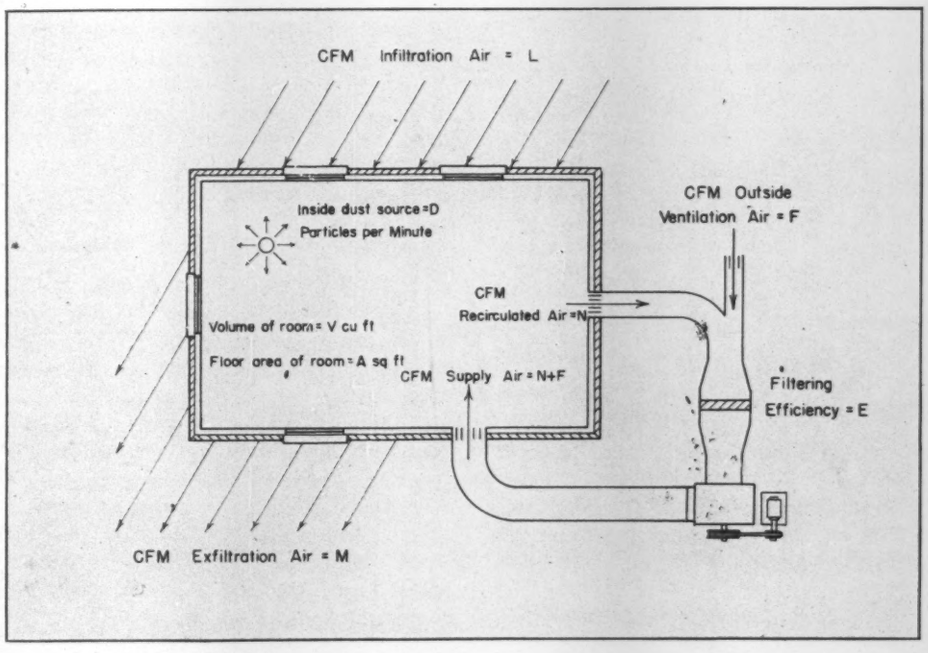
†Instructor, Engineering Experiment Station, Institute of Technology, University of Minnesota.

Fig. 1—Schematic diagram of a typical filtering installation indicating some of the terms used in the text



t = time, minutes;
C = instantaneous dust concentration, particles per cu ft;
C_o = outside air dust concentration, particles per cu ft;
C_i = initial inside air dust concentration, particles per cu ft;
C_f = final inside air dust concentration, particles per cu ft;
D = dust produced from inside sources, particles per min;
V = net volume of room, cu ft;
v = effective velocity of settling of particles, fpm;
A = floor area of enclosed space, sq ft;
H = height of room, ft;
$Q = CV$ = total number of particles in space V at time t.

Then

$$\frac{dQ}{dt} = \text{rate of change in number of particles in space } V \text{ per min} \quad [1]$$

$$= \text{rate of gain in particles per min} - \text{rate of loss in particles per min}$$

$$= V\frac{dC}{dt}$$

$$= LC_o + D + FC_o(1-E) - MC - NEC - CAv \quad [2]$$

where

LC_o = gain in particles per min by infiltration;
D = gain in particles per min by internal dust sources;
$FC_o(1-E)$ = gain in particles per min by ventilation air;
MC = loss in particles per min by exfiltration;
NEC = loss in particles per min by filtration;
CAv = loss in particles per min by settling (see December article).

If we consider V, L, C_o, D, F, E, M, N, A and v as constants and integrate between C_i and C_f and from 0 to t, then

$$\int_{C_i}^{C_f} \frac{V\,dC}{LC_o+D+FC_o(1-E)-C(M+NE+Av)} = \int_o^t dt \quad [3]$$

from which

$$C_f = \left[C_i - \frac{LC_o+D+FC_o(1-E)}{M+NE+Av}\right] e^{-t\left(\frac{M+NE+Av}{V}\right)} + \frac{LC_o+D+FC_o(1-E)}{M+NE+Av} \quad [4]$$

Equation 4 is the general dust concentration equation to be used for the prediction of dust levels under conditions of constant filtering. Specialized equations to be used for specific problems can be obtained from [4] by eliminating the unnecessary terms as in the following examples:

If infiltration equals exfiltration ($L = M$), no outside ventilation air is taken in ($F = 0$), and there is no inside dust source ($D = 0$):

$$C_f = \left[C_i - \frac{LC_o}{L+NE+Av}\right] e^{-t\left(\frac{L+NE+Av}{V}\right)} + \frac{LC_o}{L+NE+Av} \quad [5]$$

If ventilation air is taken in and a positive pressure built up in room to prevent infiltration ($L = 0$), ($F = M$), and there is no inside dust source ($D = 0$):

$$C_f = \left[C_i - \frac{FC_o(1-E)}{F+NE+Av}\right] e^{-t\left(\frac{F+NE+Av}{V}\right)} + \frac{FC_o(1-E)}{F+NE+Av} \quad [6]$$

Similar equations can be obtained in the same manner for other specific cases.

The steady flow, equilibrium condition is approached in all cases when $t \to \infty$. For example, consider the general dust concentration equation, [5]. If $t \to \infty$, then

$$\lim_{t\to\infty} e^{-t\left(\frac{M+NE+Av}{V}\right)} = 0$$

and

$$C_f = \frac{LC_o + D + FC_o(1-E)}{M+NE+Av} \quad [7]$$

This is the interior dust concentration after the filtering system has been in operation an infinite period of time. The numerator of the expression, $[L\,C_o + D + F\,C_o\,(1 - E)]$, represents the total number of dust particles entering the space per unit of time. The denominator, $(M + NE + Av)$, when multiplied by the equilibrium concentration, C_f, represents the total number of particles removed from the space per unit of time.

The physical significance of the remaining terms of Equation 4 are as follows: The quantity

$$C_i - \frac{LC_o + D + FC_o(1-E)}{M+NE+Av}$$

represents the difference between the initial dust concentration and the final concentration after equilibrium is reached. The term

$$e^{-t\left(\frac{M+NE+Av}{V}\right)}$$

is a time reduction factor. The greater the value of the fraction $[(M + NE + Av)/V]$, the shorter is the time necessary to change from the initial concentration to the final equilibrium concentration. This fraction in effect represents the number of cubic feet of the air-dust mixture removed per unit of time per unit volume of the enclosed space.

Theoretically, it requires an infinite period of time for the dust concentration to be changed from the initial to the equilibrium value as represented by Equation 7. Practically, however, the equilibrium concentration is approached in a comparatively short period of time. Fig. 2 shows the relationship between time of operation and the percentage reduction between the initial and the equilibrium dust concentrations for values of $[(M + NE + Av)/V]$ ranging from 0.002 to 0.333. The 0 per cent ordinate represents the initial concentration at the beginning of operation and the 100 per cent ordinate represents the final theoretical equilibrium concentration. The curves have been drawn only to a reduction value of 95 per cent, but if continued would all approach the axis asymptotically. A great many filtering systems have values of $[(M + NE + Av)/V]$ in the neighborhood of 0.03 to 0.04 and thus require about 100 to 200 min for the dust concentration to be reduced 95 per cent of the maximum possible reduction.

An Example

An office building has inside dimensions of 100 x 30 x 12 ft with 5 per cent of the inside air volume displaced by furniture and occupants. The air conditioning system filters and supplies to the conditioned space 720 cfm of outdoor air and 1440 cfm of recirculated air. There is 145 cfm of outdoor air introduced by infiltration. The outdoor air has a dust concentration level of 500,000 particles per cu ft; there are no inside dust sources; and the filtering efficiency based on particle count[1] is 40 per cent. Determine the interior dust concentration level under equilibrium conditions and the length of time required to lower the inside concentration 95 per cent of

[1] This same filter would be likely to have an efficiency of 80 per cent or more based on weight of dust eliminated.

the difference between the outdoor concentration and the inside equilibrium concentration.

Solution: In the solution of this problem it is first necessary to assume some typical size frequency distribution for the dust in order to evaluate the dust settling factor. Such a size distribution was determined by the U. S. Public Health Service[2] incidental to studies conducted in 14 large American cities. Table 1 shows a summary of these data together with the settling velocities which have been calculated by Stokes' Law and Cunningham's correction factor assuming spherical particles and a specific gravity of 2.0. If this distribution is assumed for purposes of the present problem, the theoretical velocity of fall may be calculated by means of Equation 2 on p. 702 of the December HPAC and results in $v = 0.0072$ fpm. The actual effective velocity of fall will be reduced from this value by turbulence and irregular particle shape. Anderson[3] has attempted to correlate the theoretical and actual velocities of fall of dusts under conditions of greater turbulence than are usually involved in conditioned rooms and recommends the use of Stokes' Law with a correction factor of 0.5. If it is assumed that this correction factor holds for the present case, the actual velocity of fall is then $v = 0.0072 \times 0.5 = 0.0036$ fpm.

The interior equilibrium may now be calculated by Equation 7. In this example $L = 145$ cfm, $D = 0$, $F = 720$ cfm, $M = 865$ cfm, $N = 1440$ cfm, $A = 3000$ sq ft, $v = 0.0036$ fpm, $E = 40$ per cent, and $C_o = 500{,}000$ particles per cu ft. Substituting these values in Equation 7, $C_f = 199{,}000$ particles per cu ft.

The time required to reduce the interior dust concentration 95 per cent of the difference between the initial and equilibrium concentrations or to a value of 214,000 particles per cu ft, is dependent upon the value of the factor $[(M + NE + Av)/V]$ (see Equation 4). In this case $(M + NE + Av) = 1452$ and $V = 36{,}000 \times 0.95 = 34{,}200$ cu ft. Hence $[(M + NE + Av)/V] = 0.042$, and from Fig. 2 the required time is 74 min.

It is interesting to note that the value of the dust settling factor, Av, is 10.8 cfm, while the value of the entire factor $(M + NE + Av)$ is 1452 cfm. This indicates that when the analysis is placed on a particle count basis the settling factor and any assumptions necessary in its evaluation, such as the application of the 0.5 correction factor to the settling velocity, are relatively unimportant.

[This is the second of the Rowley-Jordan articles on predicting dust concentration. The first appeared in the December, 1940, HPAC, and was devoted to factors affecting dust concentration levels in enclosed spaces. Future articles will cover intermittent filtering and will give typical examples of dust concentration analyses.]

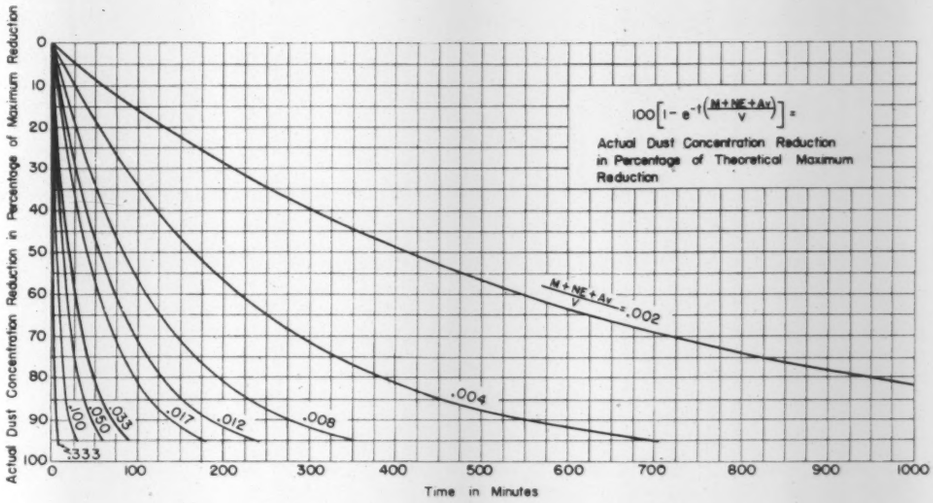


Fig. 2—Relationship between time of filtering and reduction in dust concentration

[2]Atmospheric Pollution of American Cities for the Years 1931-33, by J. E. Ives, R. H. Britten, D. W. Armstrong, W. A. Gill, and F. H. Goldman. U. S. Public Health Bulletin No. 224, p. 30, March, 1936. For similar data see also: Size Frequency of Industrial Dusts, by J. J. Bloomfield, U. S. Public Health Report No. 48, p. 964, August, 1933.

[3]Some Factors and Principles Involved in the Separation and Collection of Dust, Mist, and Fumes from Gases, by F. Anderson. Transactions, American Institute of Chemical Engineers, 16:77, 1924.

Table 1—Frequency Distribution and Calculated Settling Velocities for Atmospheric Dusts

Data from U. S. Public Health Bulletin No. 224, p. 30, March, 1936.

Diameter, Microns	Average Frequency, Per Cent	Settling Velocity, Fpm
2.0	0.34	0.0513
1.9	0.20	0.0463
1.8	0.20	0.0419
1.7	0.38	0.0372
1.6	0.33	0.0333
1.5	0.54	0.0295
1.4	0.73	0.0258
1.3	1.1	0.0224
1.2	1.8	0.0193
1.1	2.1	0.0164
1.0	4.5	0.0137
0.9	5.5	0.0113
0.8	9.5	0.0094
0.7	13.9	0.0071
0.6	19.2	0.0055
0.5	20.4	0.0040
0.4	13.1	0.0027
0.3	5.1	0.0017
0.2 or less	0.95	0.0008

CIVILIAN DEFENSE IN TIME OF WAR

Secretary of War Henry L. Stimson announced last month the appointment of a committee of seven engineers, each representing a national engineering organization, to assist the war department in technical matters relating to the collection, evaluation and dissemination of information of value in the protection of civilians and vital civilian installations in time of war.

The members of the committee, known as the National Technological Civil Protection Committee, are: Walter D. Binger, American Society of Civil Engineers, chairman; W. H. Carrier, American Society of Heating and Ventilating Engineers; Harry E. Jordan, American Waterworks Association; A. B. Ray, American Institute of Chemical Engineers; Abel Wolman, American Public Health Association; James L. Walsh, American Society of Mechanical Engineers; and Scott Turner, American Institute of Mining and Metallurgical Engineers. Major Eugene W. Ridings of the war department general staff has been designated to act as the war department contact member with the committee.

Among the problems to be studied will be those pertaining to bomb shelters, water supply and power.

"Open for Discussion"

Communications to the Editor

Panel Heating

THE EDITOR—

In reality panel heating is a cooling system—yes, a system that controls the rate of cooling of the human body. How can that be possible, you ask? Quite simple, if we but think about it. In grade school we learned that the body when perfectly healthy is at a temperature of 98 degrees. It's kept that way by the food we eat. Every mouthful of food is loaded with energy that is turned into heat. Only a small part of that heat is needed to keep the body alive; the rest must be given up. How? Well, in four ways: By evaporation, convection, conduction, and by radiation. Since the *greater* part of the body heat is surrendered *normally* by *radiation* it is that method with which we must be concerned.

For those interested in panel heating and who do not have a fundamental understanding of the principle of heat exchange by radiation, a brief explanation is in order. We know that heat always moves, flows or travels from the warmer object or surface to the colder one. Heat exchange by radiation does likewise, only it moves in the form of invisible infrared rays through space until the rays impinge on the surface of a wall or object. Heat in this form does not heat the air as the energy moves through it. The air in turn is heated by convection as it wipes either the radiating surface or the surface receiving the radiated energy. This explains why air may be very low in temperature in a room thus heated. If the walls of a room have a high enough surface temperature so that the heat from our bodies does not radiate so rapidly that we feel cold it makes little difference what the air temperature of the room may be. The air will in time assume about the same temperature as the walls unless the air is constantly changed because of ventilating needs and such air is not heated upon being supplied. Since the flow of heat is from the body to the surfaces of the spaces providing comfort it is therefore desirable to control the temperature of those surfaces so the body does not lose heat so fast that it becomes uncomfortable.

Since the control of the room surface temperatures is all important, it is essential to know what those surface temperatures should be as well as how they are to be maintained that way.

The learned Dr. Geisecke has presented formulas for determining surface temperatures for given heat losses and room air temperatures, in his articles in HPAC. He has also established rules for selecting panel temperatures, particularly if only one surface (out of a possible six) is used. It is emphatically pointed out that unless each job is carefully engineered and calculation made according to Dr. Geisecke's method or others equally good that the system just won't give full satisfaction under all conditions.

Perfect engineering can go wrong if poor judgment is exercised in the mechanical application. If the panel used as a radiating surface is constructed in a way that it has a high thermal capacity, then automatic temperature control of the condition in the space is difficult, for the residual heat energy continues to radiate long after controls indicate the desired condition has been secured. This can cause great discomfort, particularly on mild days—to say nothing of wasting fuel. Therefore, an important mechanical requisite seems to be that panels be constructed so as to have low thermal capacity.

Panels that can be fastened to or set in ceilings or walls have been developed. One type has been made by casting small diameter, thin wall copper tubes in a low density, high strength insulating material. The insulation prevents heat flow in the direction where it isn't needed. Being light weight, they are easily handled and require no more support than by screws. The tubes are cast near the surface and so spaced to give the same temperature over the entire panel area. The panels are also given a coat of material high in thermal conductivity in order to improve the radiating surface; this is done, of course, only on the radiating side of the panel. The panels have proved low in cost and high in efficiency.

Of course, no system is better than its controls. Individual room temperatures are controlled by shielded air thermostats set, of course, to function at design air temperatures. Incidentally, air temperatures (experiment indicates) as low as 50 degrees are okay; this can contribute further to fuel savings. The temperature of the fluid circulated in the panels should be controlled by a master inside-outside differential control, preferably the type sensitive to the effect of wind as well as temperature. While conventional water boilers will work, close control of fluid temperature is obtained by the instantaneous type of heat exchanger.

The time is fast approaching when all surfaces of a room will be used and then panel cooling for summer comfort will be here. A dewpoint control device will operate with the thermostat on this cycle.

Just remember, to understand panel heating, it is not the function of warming up anybody or any object. Panel heating just neutralizes and controls body heat losses by radiation, by affecting the rate of radiation from the body to the surfaces. It controls the rate at which our bodies cool or lose heat. If our bodies didn't lose heat we'd all go up in smoke—the body just has to lose heat, but of course not too fast.

—CARL F. BOESTER, Director, Housing Research, Purdue University.

Readers of HEATING, PIPING AND AIR CONDITIONING are invited to contribute their views to these "Open for Discussion" pages, which will appear from time to time. Comments on articles we have published, expressions of timely interest on developments in design, installation, operation or maintenance of heating, piping or air conditioning systems, other remarks of value to HPAC's readers—all will be welcomed by THE EDITOR.

HPAC's Annual
AIR CONDITIONING REVIEW

COMPARATIVE data on the number of installations and the connected horsepower for air conditioning commercial, industrial and residential buildings for the full year of 1939, the first 11 months of 1940, and the totals as of December 1, 1940, are tabulated on the next page. Institutional buildings and apartments are included with the commercial jobs. These figures were obtained by HEATING, PIPING AND AIR CONDITIONING from the electric utility companies serving the various areas covered, and the courtesy of these companies in making the data available for presentation is gratefully acknowledged.

If the figures for all of the cities and areas listed in the table for which complete data for both 1939 and 1940 are available are totaled, it is indicated that there was a slight gain in horsepower for the 1940 jobs and a slight decrease in the number of installations. Apparently therefore the average 1940 job was larger than the average job in 1939. The 1940 figures, it should be remembered, cover only the first 11 months of the year, while the '39 data are for the 12 months of that year.

On the questionnaire form sent by HPAC to the utility companies, the question "Do you think number and capacity of installations during 1941 will exceed 1940 jobs?" was asked. Of the 35 replies received, 24 were "yes" and only 11 "no". Estimates of the percentage increase for this year compared with last ranged all the way from 5 to 75 per cent.

Data such as that given on the next page are impossible to compile with absolute accuracy, and of course are not available for every city or area. The information can be had only from electric utilities, and therefore (in many cases) the figures given do not include air conditioning installations which are not supplied electric power by the utility. Thus, many industrial installations, and installations using some other form of refrigeration or latent heat removal than electric motor driven, are not included and the horsepower figures consequently do not entirely reflect installed air conditioning capacities. While the scope and form of the data vary for each of the reporting utilities, the comparison between 1939 and 1940 for any one city or territory is believed to be entirely accurate, for both sets of figures were obtained from the same source.

It is regrettable that more statistics on 1940 air conditioning business—and estimates for 1941—are not available from such centralized sources as government departments or trade associations. The following information from two leading air conditioning and refrigeration manufacturers is, however, of pertinent value.

Definite '41 Increase

A definite increase in air conditioning business in 1941 is predicted by J. I. Lyle, president of Carrier Corp., as a result of general business expansion during 1940.

"Defense work has accelerated industry's growing need for air conditioning," he says, "but development of new materials that can be produced only in air conditioned plants has also contributed to this growth."

A trend to "black-out proof" plants of windowless construction, where air conditioning will be a basic part of the architectural plan, is also predicted by Mr. Lyle as one of the developments in 1941. "Particularly will this be true of aviation, ammunition and other types of armament plants," he says. "Developments in Europe have emphasized the need for uninterrupted operations during air raids. Industrialists even now are beginning to realize the advantages of 'black-out proof' plants, depending solely upon air conditioning for ventilation and fluorescent lighting instead of windows. It is entirely likely this trend will carry over into other types of factories and plants in 1941."

7.5 Per Cent Sales Gain

The report of York Ice Machinery Corp. for the fiscal year ended September 30, 1940, showed a gain of 7.5 per cent in sales, an increase of 20.6 per cent in unbilled orders on hand, and net earnings of $483,-

Defense work has accelerated industry's growing need for air conditioning

Scenes from "Army on Wheels," a sound motion picture recently produced by truck division, Dodge Bros. Corp., in cooperation with war department

Name of City or Territory and Approximate Population	1939								1940 (11 Months)								Totals as of December 1, 1940							
	Residential		Commercial		Industrial		Total		Residential		Commercial		Industrial		Total		Residential		Commercial		Industrial		Total	
	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp	No.	Hp
Ala.—Practically entire state served directly and indirectly by Ala. Power Co. (958,000)	14	30.78	98	1780.11	2	788	114	2598.89	8	18.5	62	2287.53	1	655.25	71	2961.28	90	251.97	467	12213.05	20	1812.5	577	14277.52
Ariz.—Phoenix (88,000)*	47	151	17	560			64	711	18	68	13	554			31	622	198	718	98	6395	5	313	301	7426
Calif.—San Diego (160,000)			3	8.03			3	8.03			1	24			1	24			9	299.03	4	270.25	13	569.28
Colo.—Pueblo (50,000)	8	2	50	65.25			58	67.25	12	3	37	57.5			49	60.5	20	5	125	372.75			145	377.75
Conn.—Conn. Light & Power Co.*	NR	NR	NR	NR	NR	NR	NR	NR	14	18.25	15	119.85	3	7.1	32	145.2	42	67.8	73	865.43	36	572.85	151	1506.08
Conn.—Hartford (168,000)			21	346.75	1	7.5	22	354.25			17	187.75			17	187.75	7	22.5	117	2161	4	142.5	128	2326
D. C.—Washington and surrounding area (830,000)*	52	60.25	262	5145	2	34.5	316	5239.75	20	42.75	201	8927.5	1	25	222	8995.25	257	367.75	1248	64993	18	590.5	1523	65951.25
Ga.—75% of state (848,000)	8	7.45	87	1984.95	3	30.43	98	2022.83	1	7.5	63	3094.22	1	2.75	65	3104.47	74	203.71	493	15666.53	19	453.23	586	16323.47
Ga.—Savannah (105,000)	9	23.08	23	258.45			32	281.53	31	11	7	171.58			38	182.58	53	64.21	63	1294.19			116	1358.4
Ill.—Chicago (3,500,000)	271	158	895	4691	21	467	1187	5316	370	243	861	8000	19	569	1250	8812	926	788	3905	78568	190	10865	5021	90221
Ill.—Dixon (11,000)	4	6.5	15	47.25			19	53.75	16	11	3	27.5			19	38.5	29	77	70	25.5			99	102.5
Ill.—Rockford (86,000)	4	6.25	17	89.25			21	95.5	7	6.83	24	35.65			31	42.48	NR	NR	NR	NR	NR	NR	NR	NR
Ind.—Indianapolis (400,000)	17	32.4	56	392.29	8	102.3	81	526.99	50	45.8	58	1615	6	1515	114	3175.8	146	205.04	367	7985.66	26	2218.3	539	10409
Iowa—Cedar Rapids (62,000)*	NR	NR	NR	NR	NR	NR	NR	NR			10	233.5			10	233.5	NR	NR	NR	NR	NR	NR	NR	1884.25
Iowa—Sioux City (90,000)	2	1.75	34	108.75	2	8.75	38	119.25	3	1.67	15	31.75			18	33.42	37	50.5	150	1994.08	2	8.75	189	2053.33
Kans.—Kansas Elec. Pow. Co., Gen. Office, Lawrence	17	30.25	82	812.5			99	842.75	10	7.5	16	102.25			26	109.75	27	37.75	98	914.75			125	952.5
Ky.—Louisville and Jefferson County (350,000)	16	16.33	63	513.21	15	135.55	94	665.09	40	38.58	73	1100.4	11	141.33	124	1280.31	NR	NR	NR	NR	NR	NR	491	6492.48
La.—Shreveport (120,000)	NR	NR	NR	NR	NR	NR	202	3410.42	13	50.33	77	1572.35			90	1622.68	NR	NR	NR	NR	NR	NR	NR	NR
Mass.—Boston (1,400,000)*			80	1787.5	13	179.75	93	1967.25			87	972.5	5	127.75	92	1100.25	48	111	926	15600	50	2110	1024	17821
Mass.—Brockton (62,000)			6	32.5			6	32.5			1	3			1	3	3	1.75	13	102.5	2	2.75	18	107
Mass.—Cambridge (111,000)			28	678	5	40	33	718	3	2	13	60	2	19	18	81	NR	NR	NR	NR	NR	NR	NR	NR
Mass.—Springfield (150,000)	19	19.25	121	1494.5	5	91.5	145	1605.2	3	1.75	22	109.5	3	27.75	28	139	NR	NR	NR	NR	NR	NR	NR	NR
Md.—Baltimore (1,160,000)	164	135.5	272	2384.5	2	33	438	2553	278	216	380	1943.5	6	206.5	664	2366	634	624	1219	19462.5	28	1039.5	1881	21126
Me.—Cumberland and York Counties (175,000)			16	233.33			16	233.33			9	18.83			9	18.83	NR	NR	NR	NR	NR	NR	NR	NR
Mich.—Detroit and vicinity (2,538,000)	63	60	268	3262	12	281	343	3603	100	70	428	3976	7	7905	535	11951	NR	NR	NR	NR	NR	NR	1985	49068
Mich.—Lansing (92,000)			2	105			2	105			3	10.5			3	10.5	3	6.5	19	325.5			22	332
Minn.—340 Towns served by Otter Tail Power Co., Fergus Falls	5	3.75	18	76			23	79.75	3	2.5	12	39			15	41.5	16	12.75	51	127			67	139.75
Minn.—Minneapolis & St. Paul and suburbs (860,000)	18	40	161	1177	15	117	194	1334	61	39	85	865	51	297	197	1201	256	346	620	6937	335	3215	1211	10498
Mo.—Joplin (37,000)	1	0.75	5	88.75			6	89.5	2	1.75	6	37			8	38.75	6	15.5	27	375	1	30	34	420.5
Mo.—Kansas City (413,000)	105	107.67	136	1721.77	5	102	246	1931.44	77	133.65	97	1033.45	3	14	177	1181.1	578	1072.37	845	17921.45	21	400	1444	19393.82
Mo.—St. Joseph (86,000)	10	6.83	5	26.75	1	6.5	16	40.08	5	2.75	3	213.5			8	216.25	43	55.75	91	1617.67	5	143	139	1816.42
Mo.—St. Louis (1,250,000)*	378	367.7	377	7649.56	11	492.26	766	8509.52	441	448.36	381	6098.29	7	267.01	829	6813.66	1707	2414.54	1876	47299.38	86	3561	3669	53274.92
Mont.—Mont. Power Co. territory (280,000)			180	450	20	70	200	520			70	140	10	20	80	160			250	590	30	90	280	680
N. C.—Carolina P. & L. Co	NR	NR	NR	NR	NR	NR	45	624	NR	NR	NR	NR	NR	NR	18	194	NR	NR	NR	NR	NR	NR	NR	NR
Neb.—Lincoln (82,000)	11	32.24	65	3348.4			76	3380.64	7	11.83	12	251			19	262.83	NR	NR	NR	NR	NR	NR	NR	NR
N. H.—Manchester territory*	67	31.67	12	206.75			79	238.42	39	8.25	5	148	1	27.5	45	183.75	196	66.61	47	560.3	7	61.75	250	688.66
N. J.—Dover (10,000)			9	101.25	1	15	10	116.25			3	15.5			3	15.5			19	169.25	1	15	20	184.25
N. J.—Pub. Ser. Elec. & Gas Co., Newark (80% of N. J. pop.)*	NR	NR	NR	NR	NR	NR	254	2112	NR	NR	NR	NR	NR	NR	298	3396	NR	NR	NR	NR	NR	NR	1322	18711
N. M.—Mesilla Valley, Las Cruces and vicinity*	NR	NR	NR	NR	NR	NR	206	51.5	NR	NR	NR	NR	NR	NR	130	32.5	NR	NR	NR	NR	NR	NR	425	106.25
N. Y.—Central New York Power Corp. (821,000)*	21	24.5	207	3094.93	25	1456.7	253	4576.13			18	206.05	2	91.5	20	297.55	21	24.5	225	3300.98	27	1548.2	273	4873.68
N. Y.—New York (Manhattan, Brooklyn, Bronx, Queens) (6,700,000)	189	175.46	1348	18267.13	32	734.69	1569	19177.28	220	161.08	958	16354.73	58	694.28	1236	17210.09	NR	NR	NR	NR	NR	NR	5438	153177.73
N. Y.—N. Y. Power and Light Corp.*	3	8	51	498			54	506	1	2.25	24	262	3	14	28	278.25	27	44	216	2763.5	12	567.75	255	3375.25
N. Y.—N. Y. State Elec. & Gas Corp. territory*			21	283.75	3	4	24	287.75			30	271	5	47	35	318	NR	NR	NR	NR	NR	NR	93	1910.75
N. Y.—Patchogue (69,000)			4	7.5			4	7.5			7	18			7	18			20	65			20	65
N. Y.—Westchester County (500,000)	24	18.2	57	593.16	9	71	90	682.36	20	25.9	30	380.5	5	26	55	432.4	109	163.35	188	5028.27	24	319	321	5510.62
Ohio—Cincinnati (460,000)	NR	NR	NR	NR	NR	NR	231	1148.86	NR	NR	NR	NR	NR	NR	471	2645.58	NR	NR	NR	NR	NR	NR	NR	NR
Ohio—Dayton & surrounding small towns (400,000)	18	20	36	282.6	2	10.05	56	312.65	3	7.28	38	275.36	13	86.13	54	368.77	93	131.58	189	3875.76	59	1086.78	341	5094.12
Ohio—Toledo (300,000)*	NR	NR	30	897	1	60	31	957	4	10	18	280	2	8	24	298	40	82	178	4319	28	797	246	5198
Okla.—Okla. Gas & Elec. Co. territory (661,000)	70	101.21	84	736.31	6	9.17	160	846.69	33	92.59	69	907.66	2	2.05	104	1002.3	400	835.47	505	8939.03	20	113.42	925	9887.92
Okla.—Pub. Ser. Co. of Okla., Tulsa	NR	NR	NR	NR	NR	NR	NR	NR	25	80.6	47	1112.1	18	359.5	90	1552.2	171	636.3	209	12208.5	34	941.3	414	13786.1
Pa.—Lawrence, Mercer, and part of Butler, Beaver, and Allegheny counties (216,000)			7	33.5	1	3	8	36.5	2	1	7	51.5	1	5.5	10	58	5	2.5	32	283	5	11.5	42	297
Pa.—Philadelphia area (2,754,000)	64	62.66	235	4818.97	15	1399.5	314	6281.13	83	79.61	284	3566.74	36	1045.91	403	4692.26	391	540.07	1293	42671.01	168	10552.01	1852	53763.09
Pa.—Pittsburgh (1,400,000)	29	21.5	151	955.5	3	20.75	183	997.75	31	36.75	142	1549	4	55.25	177	1641	201	230.25	863	15220.75	30	819.25	1094	16270.25
Pa.—York (70,000)	16	13.67	47	427	2	50	65	490.67	2	6	15	226	1	3	18	235	NR	NR	NR	NR	NR	NR	NR	NR
R. I.—Pawtucket, Woonsocket (150,000)	3	2.25	9	53.25	2	6.25	14	61.75			2	12.5	2	8.75	4	21.25	NR	NR	NR	NR	NR	NR	NR	NR
R. I.—Providence & Southern R. I. (470,000)	3	3.09	34	428.03	3	11	40	442.12	3	2	26	219.08	3	21	32	242.08	15	29.59	126	2325.01	11	41.07	152	2395.67
S. C.—Columbia (70,000)	2	6	11	205.75			13	211.75	1	3	16	1085.5	1	7.5	18	1096	NR	NR	NR	NR	NR	NR	NR	NR
Tenn.—Chattanooga and Suburbs (250,000)	NR	NR	16	403	1	15	17	418	NR	NR	16	290	3	324	19	614	NR	NR	NR	NR	NR	NR	97	2601
Texas—Beaumont (165,000)			22	359.85			22	359.85	2	9.35	11	308.99	2	25.33	15	343.67	26	125.61	152	4651.59	4	40.33	182	4817.53
Texas—El Paso (130,000)*			2	85			2	85			4	34.25			4	34.25			65	1864	11	82.5	76	1946.5
Texas—San Antonio (253,000)	19	60.67	28	398			47	458.67	10	45.75	24	432.5			34	478.25	97	272.25	207	10856	2	4	306	11132.25
Texas—Texas Electric Service Co. system, Ft. Worth (422,942)	34	76	43	2082			77	2158	17	115	32	930			49	1045	114	558	208	11087			322	11645
Va.—Charlottesville (19,000)			23	297.8			23	297.8			31	538.3			31	538.3	NR	NR	NR	NR	NR	NR	159	1738.5
Va.—Lynchburg and vicinity (50,000)	5	5	6	5.5	1	1	12	11.5	5	2.5	3	18	1	1	9	21.5	11	8.5	16	51	3	3.5	30	63
Va.—Va. Electric & Power Co., Richmond	3	1	84	1380	3	495	90	1876			45	961	5	475	50	1436	NR	NR	NR	NR	NR	NR	463	12461
W. Va.—Northern W. Va. (Monongahela System)	6	6	10	140			16	146	4	3	10	49			14	52	16	15	43	525			59	540
Wis.—Milwaukee (590,000)	14	8.7	81	703	6	41	101	752.7	19	14	97	516	4	68	120	598	65	102	371	7630	45	1374	481	9106
Wis.—Superior (35,000)			7	59	1	1.5	8	60.5			3	3.5			3	3.5			10	62.5	1	1.5	11	64
Wis.—Wis. P. S. Co. (350,000)	3	5	4	20	6	75	13	100	12	20	8	40	3	50	23	110	NR	NR	NR	NR	NR	NR	NR	NR

*See footnote on following page. "NR" means no record.

122 after taxes, contrasted with a loss of $185,077 the year before.

In the company's letter to stockholders higher volume and better control over costs were credited for the improvement in profits which contributed to a stronger position in both cash and net current assets. With sales of $16,163,895 only $1,133,617 greater than in the preceding year, net earnings before income taxes showed an improvement of $782,682. Working capital increased by $566,792 and the company closed its year with a larger cash balance on hand than at any year-end during the past decade.

The program of research and development carried on has greatly broadened the corporation's sales outlets and transformed the nature of its business. The report pointed out that outside of supplies and repairs "about 80 per cent of the underlying products forming the nucleus of sales in recent months have been developed during the past eight years."

The corporation's part in the national defense program was cited as illustration of the diversified nature of its activities. Typical of recent business in this connection were "refrgeration and air conditioning products for modern windowless industrial plants, smokeless powder plants, machine tool and other precision instrument factories, a variety of plants manufacturing synthetic products, army cantonments, naval bases and a wide range of naval and merchant marine ship requirements."

Footnotes

The following footnotes pertain to the cities or territories marked with an asterisk (*) in the table on the preceding page:

Ariz.—Phoenix: There are approximately 19,500 evaporative coolers representing an estimated 3200 hp in addition to the air conditioning installations tabulated.

Conn.—Conn. Light & Power Co.: Figures given include Bristol, Danielson, Essex, Greenwich, Meriden, New Britain, New Milford, Northern, Norwalk, Waterbury, Willimantic, Winsted.

D. C.—Washington: Commercial installations include government buildings.

Iowa—Cedar Rapids: Room coolers not included in data.

Mass.—Boston: 1940 data cover first 10 months only, and totals are as of November 1, 1940. Figures do not include 367 room coolers of 1 hp or less totaling 274 hp.

Mich.—Lansing: There are 780 tons capacity in jobs using well water, which is available at 55 F.

Mo.—St. Louis: 1939 figures include first 11 months only, rather than the entire year.

N. H.—Manchester territory: Practically all the residential installations included are winter air conditioning.

N. J.—Pub. Serv. Gas & Elec. Co.: Figures do not include industrial process air conditioning.

N. M.: These are all evaporative cooling installations.

N. Y.—Central N. Y. Power Corp.: Figures cover Syracuse, Oneida, Rome, Utica, Herkimer, Watertown, Ogdensburg, Malone, Potsdam, Oswego, Fulton and Cortland.

N. Y.—N. Y. Power & Light Corp.: Figures cover Albany, Troy, Schenectady, Hudson, Glens Falls, Saratoga, Warrensburg, Gloversville, Amsterdam, Canajoharie, Cobleskill, Greenwich.

N. Y.—N. Y. State Elec. & Gas Corp. territory: Figures cover parts of Clinton, Washington, Rensselaer, Columbia, Dutchess, Putnam, Westchester, Chautauqua, Cattaraugus, Erie, Wyoming, Niagara and Orleans counties, and all territory south of the New York Central not covered by the Central New York Power Corp., except Rochester Gas.

Ohio—Toledo: Totals for 1939 and as of December 1, 1940, do not include residential installations.

Tex.—El Paso: Figures do not include 4851 evaporative cooling installations now on El Paso Electric Co. lines.

Gas Air Conditioning

Last year was one of orderly progress in the field of gas summer air conditioning, according to Charles R. Bellamy, chairman of the American Gas Association's joint committee on industrial and commercial summer air conditioning in the committee's 1940 report. Its most encouraging feature was not any startling new discovery or development, but a growing realization on the part of gas company executives of the real status of gas summer air conditioning equipment.

Manufacturers of gas air conditioning equipment improved their offerings or obtained valuable additional field test information. This work resulted in new installations of gas summer air conditioning equipment and clarified the lines along which future equipment development should take place.

One company made a number of improvements in its line of continuous rotary silica gel dehumidifying units which were originally introduced in the spring of 1939. In addition, this organization expanded its line of rotary equipment by the introduction of an 800 cfm unit to supplement the 500 and 1300 cfm units previously offered. A number of interesting applications of these units and the older and larger stationary bed units offered by this company have been made, caring for dehumidification on comfort air conditioning installations or to meet special industrial drying or low humidity requirements.

Another company placed 15 or more ammonia absorption refrigeration systems of a unique new design in experimental field operation. This system employs a direct fired gas generator and a novel thermal transfer device for circulating fluids within the sealed refrigeration unit. The unit includes an evaporative condenser as an integral part. While the present experimental unit has a capacity of only 2½ tons, the design provides for the storage of additional refrigeration effect to meet higher peak load requirements by freezing ice on the evaporator coils. The unit is designed for location outside of the building in which the refrigeration effect will be utilized.

Another manufacturer continued to offer equipment using activated alumina as a direct dehumidifying agent, designed to meet special industrial or commercial applications. While much of this equipment is used for the dehumidification of gases at high pressure in closed systems and employs steam or electrical heat for reactivation purposes, direct gas fired dehumidifiers for special or general applications can be provided.

Another manufacturer continued development of a unit for year 'round air conditioning of a size suitable for residential use, which provides summer cooling and dehumidifying by means of a gas fired absorption refrigeration system using water as the refrigerant and lithium chloride solution as the absorbent. The system is hermetically sealed and is designed to withstand the lay-over periods common to air conditioning equipment without impairment in operating efficiency or physical condition of the apparatus. A single unit is designed to care for the entire air conditioning requirement, providing circulation and cleaning of the air throughout the entire year and affording heating and humidifying in winter and cooling and dehumidifying during the summer. Gas is used as the major operating medium, though a conventional electric motor driven fan is used for air circulation. While the present unit requires the use of tap

[Concluded on p. 107]

Heating Compressed Air Piping

Electric Units Prevent Ice Formation and Freeze-Ups . . . By Lee P. Hynes*

SUMMARY—It is often necessary to heat compressed air lines or reservoirs, to prevent freezing or to supply hot process air. Electricity can be used, but each application presents problems which must be carefully solved. . . . . An example of this use of electric heat is in connection with air bubbler systems for preventing ice formations along the faces of dams, gates and other hydraulic structures. Studies by D. F. McFarland on installations in Maine indicate that ice troubles at the nozzles of these bubbler systems can be prevented by introducing approximately 300 watts of electric heat into each air feeder branch pipe and that the heat should be applied where the air pipe enters the surface of the water. Unless heat is applied, a hollow ball of ice was found to form around the nozzle, completely blocking the air flow. . . . . In using electric heat for heating compressed air lines, surface density of the heater must be low enough to avoid burn-outs under any condition, says Mr. Hynes, and protection against igniting oil vapors must be assured. The electrical connections must be thoroughly protected from moisture, oil and dirt

IT is often necessary to heat compressed air lines or reservoirs, sometimes to prevent freezing of condensation and sometimes in order to obtain hot air for an industrial process. Electricity can be successfully employed for this purpose, but there are special problems to be solved in its application.

An example is the application of electric heat to compressed air lines for air bubbler systems. Air bubblers are used to prevent ice formation along the faces of dams, gates and other hydraulic structures, the compressed air being discharged from jets at a point preferably about 20 ft below the surface of the water. The rising air bubbles induce an upflowing column of comparatively warm water, thus preventing the formation of ice on the surface of the pond or stream.

It is customary to install an air compressor at some convenient location and run a pipe line along the dam or other structure with branch pipes leading down into the water. These branch pipes terminate in headers with small orifices to discharge the air into the water. This method is effective but has usually been carried out by rule-of-thumb methods. Considerable trouble has been experienced due to lack of proper design data because little has been known about what really takes place at the air outlets under different operating conditions.

In very cold weather the air becomes chilled while passing through exposed pipe lines and often reaches a temperature well below zero. When this very cold air flows down through a pipe entering the water it chills the pipe and freezes a heavy coating of ice around it. This will be heaviest at the water line and will taper down in thickness as the pipe extends deeper into the water because some of the "cold" is "absorbed" in the process of freezing ice around the pipe.

Frequently the air will reach the nozzle orifices still cold enough to freeze ice over them and block the air flow. This renders the system inoperative and the feeder pipes must frequently be raised out of the water so that the ice can be cleaned off. Unless this is done quickly ice will freeze on the water surface and require chopping. This involves a heavy expense for labor, and is a cold and disagreeable job for workmen.

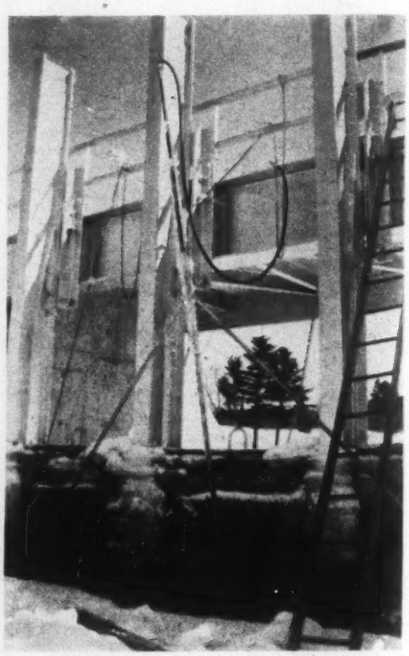
D. F. McFarland, an industrial electrical engineer, of Bingham, Me., has made a careful study of bubbler systems for ice prevention and has obtained some very interesting and useful data. He made studies of various types of air nozzles and of methods for equalizing the air flow to all the nozzles at the most effective pressures. He also designed better mechanical arrangements for supporting the jets under water so that they can easily be lifted up for inspection or adjustment.

For the best operating results he advises an air pressure just slightly in excess of the water pressure due to the head at the nozzles. This will often be in the neighborhood of 10 lb ga. As air emerges from a nozzle it forms a large air bubble around the jet. Small bubbles constantly break loose and rise to the surface, inducing a strong upflow of water.

Application of Electric Heat

Mr. McFarland also studied the application of electric heat to air lines feeding bubbler jets and found that freeze-ups could be entirely prevented by the use of compara-

Fig. 1—View of an hydraulic gate structure where air bubblers are used to prevent the heavy ice sheet on the surface of the water from freezing solid to the structure. A narrow band of open water can be seen between the white ice in the foreground and the dark background of the vertical structure. One of the header units has been lifted up to the surface. The central branch feeder pipe carries air down to the horizontal header pipe and the diagonal pipes are simply structural braces. Spaced along the horizontal header are pipe orifices through which air is discharged into the water. The assembled unit is carried by a hoist chain and flexibility is secured by means of an air hose between the top of the feeder pipe and the stationary air supply pipe on the structure. The unit shown was not heated and ice has formed on the pipe, being heaviest at the water line



*Mechanical and Electrical Engineer. Member of HPAC's Board of Consulting and Contributing Editors.

tively small amounts of electric power if applied in the proper manner. He found that unless heat was applied the air was often cold enough to freeze the large air bubble into a hollow ball of ice around the nozzle and completely block the air flow. Some of these hollow ice balls are shown in the photographs.

Experience proved that ice troubles at nozzles could be completely prevented in Maine by introducing approximately 300 watts of electric heat into each air feeder branch pipe, and that this heat should be applied where the air pipe enters the surface of the water. Even in the coldest weather this quantity of heat warmed the flowing air sufficiently to overcome all

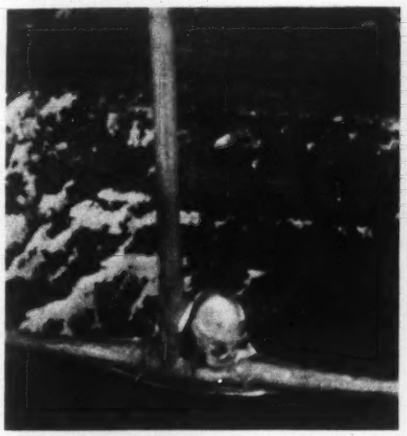


Fig. 2—View of the junction between the vertical feeder pipe and the horizontal header pipe. One orifice is located close to this junction and a hollow ice bubble is shown frozen over this jet, thus rendering it inoperative by blocking the flow

danger of freezing up the nozzles. It also prevented a heavy coating of ice on the exterior surface of the feeder pipe below the water line, and thus reduced mechanical stresses on the pipe and its supports.

In coöperation with Mr. McFarland, the writer has designed electric heating means which are compact, mechanically rugged, and waterproof so that they can be connected into the air feeder pipes at the water line. Special waterproof connections are used with heavy duty flexible rubber cable to bring the power supply down to the heater from a stationary overhead supply. Flexibility for each complete bubbler unit assembly is obtained by using a section of air hose in each branch air feeder pipe. This

Fig. 3—One of the ice bubbles lying in the palm of a workman's hand against a dark background. (This background is simply the side of an automobile and of course has no relation to the equipment under discussion in this article)

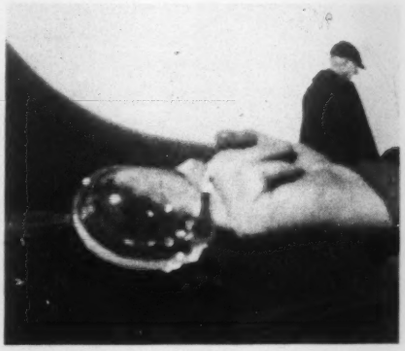


Fig. 4—Another view of one of the hollow ice bubbles. The man's two fingers are inserted inside. This also is taken against the background of an automobile

hose runs from the main air supply line to the top of the electric heater unit. The air flows through the heater on its way to the nozzles and is warmed. The electric cable can be spiralled around the air hose, which serves as a support for it.

Careful Design Essential

There are many other applications for electric heat in compressed air lines, but careful design is always essential. The surface density of the heater must be low enough to avoid burn-outs under any condition, even though air flow should cease. Because compressed air usually carries oil vapors an explosive mixture of air and oil vapor is often present and complete protection against ignition of this mixture must be assured. The electric connections must also be thoroughly protected from water, oil and dirt, and the assembled unit must be mechanically strong and light in weight.

A. C. Review—

[Concluded from p. 105]

water for condensing purposes during the summer season, the company has under development an evaporative condenser which will eliminate the present requirement for the supply and disposal of substantial quantities of tap water.

A fifth manufacturer is offering absorption dehumidification units using lithium chloride solution as the dehumidifying fluid and produced on a production basis. Two standard cell sizes with respective capacities of 3000 and 4000 cfm are available. These units can be arranged in multiple to handle installations of almost unlimited capacity. Single units or multiple assemblies can be used to provide dehumidification for commercial or industrial installations of the larger than average size class. While for practical reasons steam must be used as the direct source of heat for regeneration of the lithium chloride solution, many of the installations obtain steam from gas fired boilers.

During 1940, another concern concentrated on the modification of its absorption refrigeration system along lines which make possible the production of a sealed unit and the elimination of certain minor difficulties encountered in the field operation of prior designs. Fifteen units of this new design, each with a capacity of approximately 35 tons, have been constructed. Eight of these units have been placed in operation to provide air conditioning for a university library.

In its 1939 report, the committee presented a discussion of the sales status of summer air conditioning with gas, in which attention was given to the practical operation of sales outlets in the general air conditioning field. The situation in the general air conditioning field and the inevitable relation of the gas industry to this field did not change during the past year, according to Mr. Bellamy. An analysis of the equipment offerings described will indicate that their sale must dovetail with the sale of other air conditioning equipment, and that their installation can in many cases be most properly handled by those engineering and contracting firms making conventional air conditioning installations.

HORSEPOWER REQUIREMENTS of Refrigerating Compressors

William Goodman* Presents a Simple Method of Computing Total and Maximum Horsepower

SUMMARY—Computing the horsepower of compressors from tables of "horsepower per cubic foot" of refrigerant vapor circulated is much simpler than computing the horsepower from tables of "horsepower per ton" such as have been widely used heretofore. The use of tables of horsepower per cubic foot entirely divorces the power requirements of the compressor from its refrigerating capacity. The power required by a compressor depends only upon the compressor itself, whereas its refrigerating capacity depends upon other factors external to the compressor. . . . Furthermore, these articles clearly bring out the little-recognized fact that the horsepower of a compressor does not continue to increase steadily with an increase in suction temperature as many believe. There is one suction temperature at which the horsepower is a maximum. Above and below this one suction temperature the horsepower will be lower than the maximum. Formulas—and tables to simplify their use—are presented for computing the suction temperature at which the horsepower is a maximum and for computing the value of the maximum horsepower itself. . . . Altogether, these articles present a novel approach to the subject of computing the theoretical horsepower of compressors. This approach simplifies basic concepts and should prove useful and illuminating

THE tables of horsepower per cubic foot of vapor compressed per minute for "Freon-12" and ammonia (given on p. 712 of the December HPAC, and p. 20 of the January HPAC, respectively) were computed entirely from the thermodynamic properties of these refrigerants by conventional thermodynamic methods. The perfect gas laws were not used in computing them. However, in deriving the equations for p_e/p_m and hp_m, the perfect gas laws were used. Therefore, it is interesting to note that the results of the examples given check with the values listed in the tables. In the ammonia table, under the column for 200 lb absolute head pressure, the maximum horsepower is given as 0.3625 at an evaporator temperature of 34 F, whereas the result of the first example on p. 22 of the January article indicates that the maximum horsepower is obtained at 33.6 F, and from the second example on the same page its value is 0.3628 hp. Hence, it is evident that in spite of the fact that the equations are based on the perfect gas laws, they can be used with a reasonable degree of accuracy to predict the suction pressure at which the maximum theoretical horsepower will be obtained, and the value of the maximum horsepower itself, if the correct value of the adiabatic exponent k is known.

It is of interest to check one of the tabulated values for "Freon-12;" this is done in the following example.

Example: A "Freon-12" compressor is to operate at a head pressure of 115.3 lb ga. For "Freon-12", $k = 1.16$. Find (a) the evaporator gage pressure and temperature at which the horsepower is a maximum. (b) The maximum hp per cu ft per min.

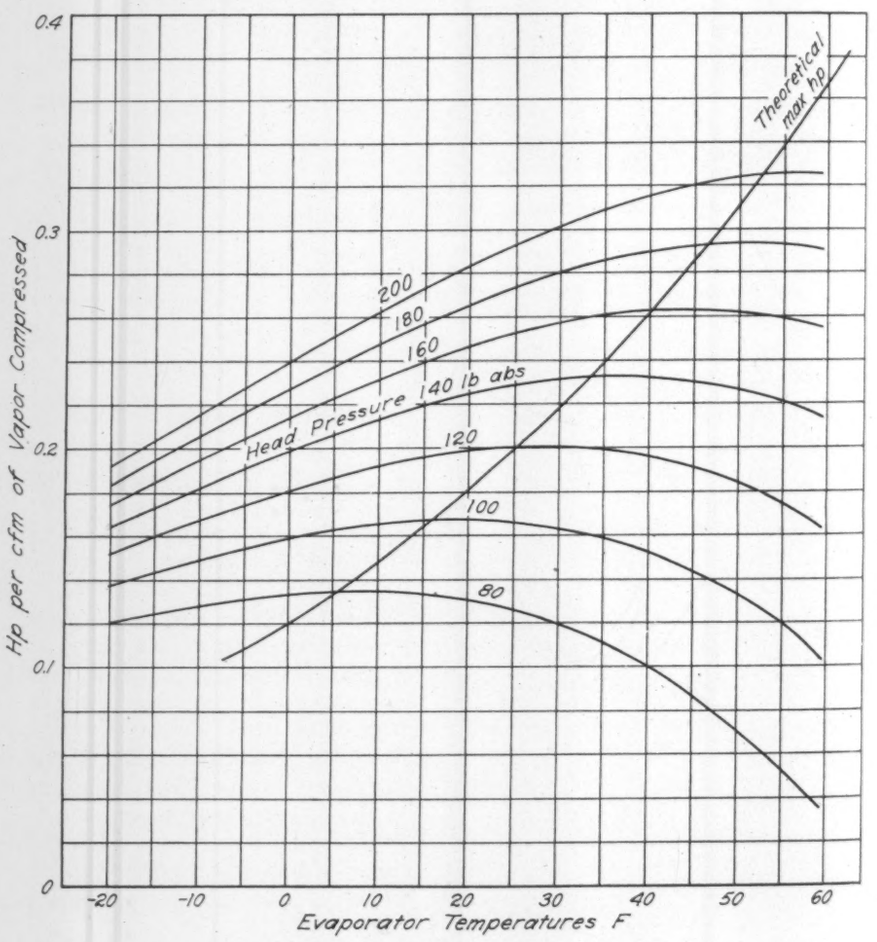


Fig. 1—Theoretical horsepower for "Freon-12"

*Consulting Engineer, The Trane Co. Member of HPAC's Board of Consulting and Contributing Editors.

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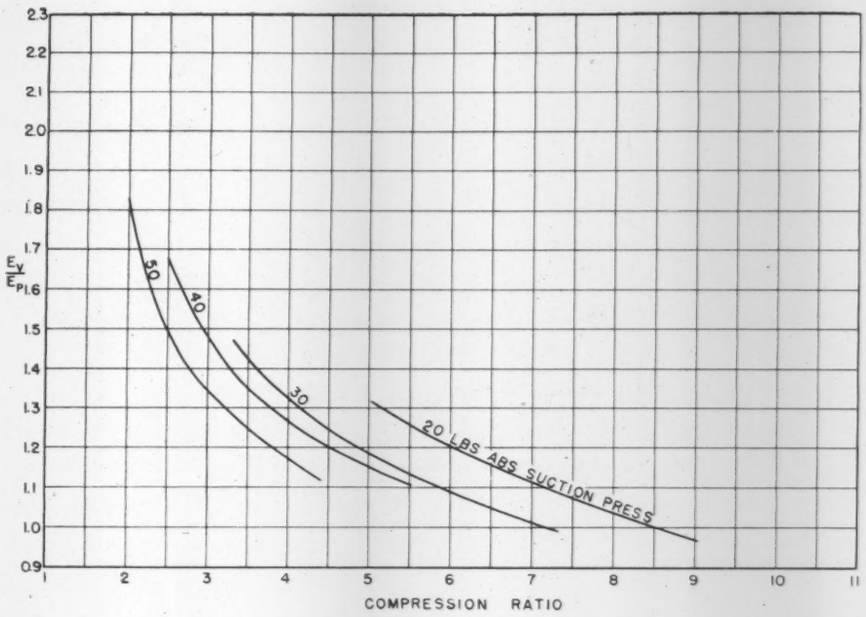

Fig. 2—Values of the ratio E_v/E_p computed from the tests of Reed and Ambrosius and of Irving on a 4½ in. x 4½ in. vertical, single acting ammonia compressor with 5.56 per cent clearance

Solution: (a) $p_c = 115.3 + 14.7 = 130$ lb ga.

Referring to column 2 of the table on p. 21 of the January article, for $k = 1.16$, $p_c/p_m = 2.933$. Therefore, $p_m = 130/2.933 = 44.3$ lb per sq in. absolute. Therefore, the gage pressure $= 44.3 - 14.7 = 29.6$ lb per sq in.

Referring to a standard table of properties of "Freon-12", the evaporator temperature at which the horsepower is a maximum is 31.4 F.

(b) From column 3 of the table on p. 21 of the January article, for $k = 1.16$, $Z = 0.001725$. Using the equation, $hp_m = Z\ p_c$, $0.001725 \times 130 = 0.2243$, the theoretical maximum hp per cu ft per min.

From the "Freon-12" table, the maximum horsepower for a head pressure of 130 lb per sq in. absolute occurs at an evaporator temperature of 32 F, whereas from the preceding example the answer is 31.4 F. Also, from the same table the value of the maximum horsepower is 0.2166 whereas from the preceding example it is 0.2243. Considering the different methods of computation, this agreement is reasonably good and indicates that the equations may also be used for "Freon-12" compressors.

As can be seen from the preceding example and the one on p. 22 of the January article, there is a larger difference between the computed and the tabulated values of the maximum theoretical horsepower for "Freon-12" than for ammonia. This larger difference is due to the fact that the individual values of the exponent k for "Freon-12" deviate by relatively large amounts from the average value of 1.16. In the case of ammonia, on the other hand, the individual values are all very close to the average value of 1.29. These statements are borne out by an examination of the tables which show individual values of the exponent k for various head pressures and suction temperatures. The computation of these tables is discussed later.

The theoretical horsepower required for "Freon-12" is shown in Fig. 1. These curves were plotted from the data in the "Freon" table and clearly show the maximum horsepower. Also shown in Fig. 1 is the curve of theoretical *maximum* horsepower computed by means of the equations based on the perfect gas laws. From Fig. 1 it is apparent that for "Freon-12," the suction temperature at which the theoretical horsepower is a maximum is slightly higher than the temperature obtained by means of a computation based on the perfect gas laws.

Maximum Brake Horsepower

In actual compressors the maximum brake horsepower may usually occur at a slightly higher suction temperature than the suction temperature computed by means of the equation for p_c/p_m. This is due to variations in the value of the ratio E_v/E_p. However, as can be seen from the table comparing three refrigerants, the suction temperature at which the brake horsepower is actually a maximum will usually not be far from the theoretical suction temperature computed by means of the equation for p_c/p_m.

Values of Ratio E_v/E_p

Whether or not the brake horsepower will be a maximum at the theoretical critical compression ratio depends upon the manner in which the ratio E_v/E_p varies. Unfortunately, there is little experimental data available from which both values of the volumetric efficiency E_v and values of the over-all power efficiency E_p can be computed. The best set of tests available are those of Reed and Ambrosius and those of Irving. [See Refrigerating Engineering, March, 1931, p. 176, and July, 1932, p. 12.] The tests of Reed and Ambrosius and of Irving were run on the same compressor at the University of Illinois—a 4½ in. x 4½ in., vertical, single acting ammonia compressor with 5.56 per cent clearance. The tests of Reed and Ambrosius gave the volumetric efficiency as a function of the compression ratio and showed that speed had no effect upon its value. The data of Irving are all given at a speed of 500 rpm. The values of

Symbols

Bhp = brake horsepower required.
E_p = over-all power efficiency, per cent, $= Thp/Bhp$.
E_v = volumetric efficiency of compressor, per cent.
hp_m = maximum theoretical hp required per cu ft of vapor compressed per minute.
k = ratio of the specific heats of the refrigerant.
p = suction pressure of vapor, lb per sq in. absolute.
p_c = condenser pressure, lb per sq in. absolute.
p_m = suction pressure for which the theoretical hp will be a maximum, lb per sq in. absolute.
Thp = theoretical horsepower required for isentropic compression.
Z = a constant.

the over-all power efficiency E_p were computed from the data of Irving, and using the values of volumetric efficiency given by Reed and Ambrosius, values of the ratio E_v/E_p given in Fig. 2 were computed. Practically all the points plotted fell exactly on the curves shown. However, the curve for a suction pressure of 20 lb absolute does not seem to be properly located in relation to the three curves for the higher pressures. Nevertheless, the data in Fig. 2 give the results of the best experimental data so far available.

For convenience in computations, Fig. 2 shows values of the ratio E_v/E_p. As a matter of interest, a better test of the location of the curves is obtained by plotting the reciprocals; in other words, values of E_p/E_v. On such a plot all of the curves should run through $E_p/E_v = 0$ for $p_c/p = 1$. The reason for this can be found by analyzing the values of E_v and E_p for a compression ratio of one. As stated by Reed and Ambrosius for p_c/p, $E_v = 1$. This is borne out by their tests. On the other hand, for a compression ratio of one, the theoretical horsepower is zero. Inasmuch as E_p is the ratio of the theoretical to the actual horsepower, it is evident that $E_p = 0$ for $p_c/p = 1$. Hence, for $p_c/p = 1$, curves for the ratio E_p/E_v should run through zero. If the reciprocals of the values of E_v/E_p used in drawing Fig. 2 are plotted, the extensions of the curves will be found to pass through zero for $p_c/p = 1$.

Values of brake horsepower computed by using the data in Fig. 2 will be higher than are generally used in commercial work. The data in Fig. 2 are based on the results of Irving's tests, and these tests gave rather high values of brake horsepower.

Values of Exponent k for "Freon-12"

Suction Temperature F	Head Pressure, Lb per Sq In. Absolute							
	80	100	120	140	160	180	200	220
	Condensing Temperature, F							
	66.3	80.9	93.4	104.5	114.5	123.7	132.1	139.9
—20	1.150	1.150	1.151	1.152	1.152	1.153	1.154	1.156
—10	1.150	1.151	1.152	1.153	1.154	1.155	1.156	1.157
0	1.152	1.152	1.153	1.154	1.155	1.156	1.157	1.159
10	1.153	1.153	1.154	1.156	1.156	1.158	1.159	1.160
20	1.155	1.155	1.156	1.157	1.158	1.159	1.161	1.162
30	1.154	1.156	1.157	1.159	1.160	1.161	1.163	1.165
40	1.157	1.158	1.158	1.160	1.161	1.163	1.166	1.167
50	1.159	1.158	1.160	1.162	1.163	1.166	1.168	1.169

Average of all values = 1.157.

Values of Exponent k for Ammonia

Suction Temperature F	Head Pressure, Lb per Sq In. Absolute				
	160	180	200	220	240
	Condensing Temperature, F				
	82.6	89.8	96.3	102.4	108.1
—20	1.297	1.295	1.294	1.293	1.293
—10	1.297	1.296	1.295	1.294	1.293
0	1.297	1.296	1.295	1.295	1.294
10	1.297	1.296	1.296	1.295	1.294
20	1.296	1.295	1.295	1.295	1.294
30	1.296	1.296	1.294	1.294	1.293
40	1.294	1.293	1.294	1.293	1.293
50	1.291	1.292	1.292	1.293	1.292

Average of all values = 1.294.

Considering the utility of the ratio E_v/E_p for accurately computing the brake horsepower of compressors, it is highly desirable that further investigations should be undertaken to determine its value for typical commercial compressors under a wide variety of operating conditions.

Values of k

The value of k is of importance in determining the critical compression ratio and the maximum theoretical horsepower. Therefore, it is desirable that the value of k for the commonly used refrigerants should be known as accurately as possible.

For this reason, a series of computations were made of the final temperature after isentropic compression through the ranges of head pressures and suction temperatures listed in the tables given here. These computations were made by using the published tables of the thermodynamic properties of "Freon-12" and ammonia. The actual thermodynamic properties of the refrigerants were used for these computations; no assumptions involving the perfect gas laws were used.

In the tables shown here, the computed value of k is given for each combination of suction temperature and discharge pressure listed. The average rounded values of k obtained in this manner are 1.16 for "Freon-12" and 1.29 for ammonia.

[This is the third of three articles on this subject. The others appeared in HPAC for December, 1940, and January, 1941.]

Comparison of Three Refrigerants

Refrigerant	Condensing Pressure for Curves of Figure[a] Lb per Sq In. Abs	k	Critical Compression Ratio from Table[b]	Actual Suction Temperature (Approx.) at Which Brake Horsepower is a Maximum as Read from Figure[a] F	Theoretical Suction Temperature at Which Theoretical Horsepower is a Maximum as Computed from Equation F
"Freon-12"	125.5	1.16	2.933	34	29.5
Ammonia	199.7	1.29	3.104	38	33.5
Carbon dioxide	1030.0	1.30[c]	3.117	7	4.8

[a] See figure on p. 713, December, 1940, HPAC.
[b] See table on p. 21, January, 1940, HPAC.
[c] Perry, Chemical Engineers' Handbook, p. 487.

For news of new and improved products, see Equipment Developments (back section) each month in HPAC.

Heating·Piping·Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

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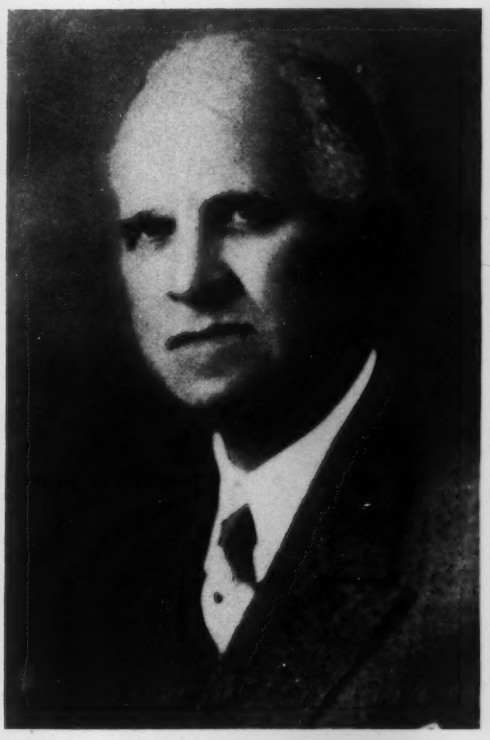

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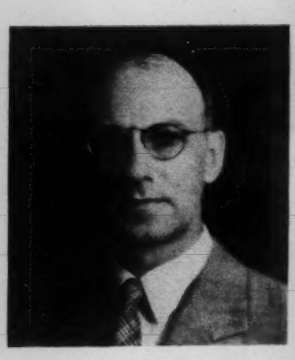

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Friction-Heads in Standard Six-Inch Pipe

By F. E. Giesecke* and J. S. Hopper**

College Station, Tex.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Agricultural and Mechanical College of Texas, and conducted at the Engineering Experiment Station.

SUMMARY—In order to obtain friction-heads for pipe of the larger sizes used in forced circulation hot water heating systems a new series of tests on 6-in. standard black pipe was inaugurated. The use of screwed couplings, welded joints and elbows was studied, and the results showed that friction-heads in couplings and in well fabricated welds can be neglected in friction-head calculations. A method of using a welded elbow as a flow meter is outlined and the investigation indicated the necessity of additional research to determine the effect of water temperature and the effect of incrustation resulting from corrosion on the friction-head in pipes and fittings.

Introduction

THE chart of friction-heads in black iron pipe, published in the HEATING, VENTILATING, AIR CONDITIONING GUIDE of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, is based on previous experimental determinations[1]. In these earliest experiments determinations were limited to pipes ranging in size from ½-in. to 3-in. The friction-heads per foot of pipe for these sizes were the following:

For—

½-in. pipe $h = 0.014335v^{1.776}$ ....(1)

¾-in. pipe $h = 0.00855v^{1.812}$ .....(2)

1 -in. pipe $h = 0.006371v^{1.7707}$ ....(3)

1¼-in. pipe $h = 0.004675v^{1.756}$ .....(4)

1½-in. pipe $h = 0.003228v^{1.7068}$ ....(5)

2 -in. pipe $h = 0.00253v^{1.6787}$ ....(6)

2½-in. pipe $h = 0.00253v^{1.735}$ .....(7)

3 -in. pipe $h = 0.00166v^{1.679}$ .....(8)

By plotting the coefficients and exponents of v in these eight equations on logarithmic paper, it was found that they could be combined into the following:

$$h = 0.00685 \frac{v^{1.79d^{-0.04}}}{d^{1.275}} \quad \ldots\ldots\ldots\ldots (9)$$

Where

h = the friction-head in feet of 70 F water, when 70 F water is flowing in standard black pipe ranging in size from ½ in. to 3 in.

v = velocity of water in feet per second.

d = actual internal diameter of pipe in inches.

In order to apply this equation to the design of hot-water heating systems, tests were made to determine how the friction-head is affected by the temperature of the water, and it was found[2] that it varies approximately as $t^{-0.19}$.

Equation (9) may therefore be changed to

$$h = \frac{0.01533v^{1.79d^{-0.04}}}{t^{0.19}\, d^{1.275}} \quad \ldots\ldots\ldots\ldots (10)$$

in which t represents the temperature of the water, in degrees Fahrenheit, flowing in the pipe and in which h is expressed in feet of water of the same temperature.

If the average temperature of the water in a hot water heating system is 140 F, the friction-heads in the pipe lines of the system are

$$h = \frac{0.006v^{1.79d^{-0.04}}}{d^{1.275}} \quad \ldots\ldots\ldots\ldots (11)$$

so long as the pipes are new and clean.

In order to determine how the friction-head in the pipe lines of a hot-water heating system may be expected to increase during use, some pipes which had been in continuous use for about seven years were taken from the hot-water heating system of the university greenhouse; and it was found that the coefficient of v in Equation (9) was about 17 per cent greater and the exponent about 6 per cent greater than in new and clean pipe.

Since the average temperature of water in a hot water heating system is generally somewhat higher than 140 F, the coefficient of v in Equation (11) was increased by 15 per cent (instead of 17 per cent) and the exponent by 5 per cent (instead of 6 per cent) producing the following:

$$h = 0.0069 \frac{v^{1.88d^{-0.04}}}{d^{1.275}} \quad \ldots\ldots\ldots\ldots (12)$$

This equation was used to design the charts in THE GUIDE, 1940. The experimental data were determined for pipes ranging in size from ½ in. to 3 in. but the results seemed so consistent that it was considered safe to apply the resulting equation to pipes ranging in size up to 6 in.

When the chart was designed, hot water heating was not very popular and was applied, as a rule, only to small installations in which no large pipes were used. Since then, conditions have changed materially and many of the new heating installations are for hot water. Many installations are of considerable magnitude, requiring the use of large pipes, so it is important to know whether the charts in THE GUIDE, which are based on tests of small pipes, are sufficiently accurate to be used in the design of systems involving large pipes. Therefore, a Technical Advisory Committee† of the Society was organized and the program outlined provided for experimental determination of the loss of head in 6-in. standard iron pipe which is described in this paper.

Objects

The principal object of the investigation was to determine the friction-heads, for water of ordinary temperature, in:

1. New and clean standard 6-in. black pipe.
2. Screwed couplings of 6-in. pipe.
3. Welded joints of 6-in. pipe.

*Professor Emeritus, Heating, Ventilating and Air Conditioning, A and M College of Texas. MEMBER of ASHVE.

**Assistant Professor of Mechanical Engineering, A and M College of Texas. MEMBER of ASHVE.

[1]The Friction of Water in Pipes and Fittings, by F. E. Giesecke. (*Bulletin* No. 1759, University of Texas.)

Presented at the 47th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., January, 1941.

[2]Loc. Cit. See Note 1.

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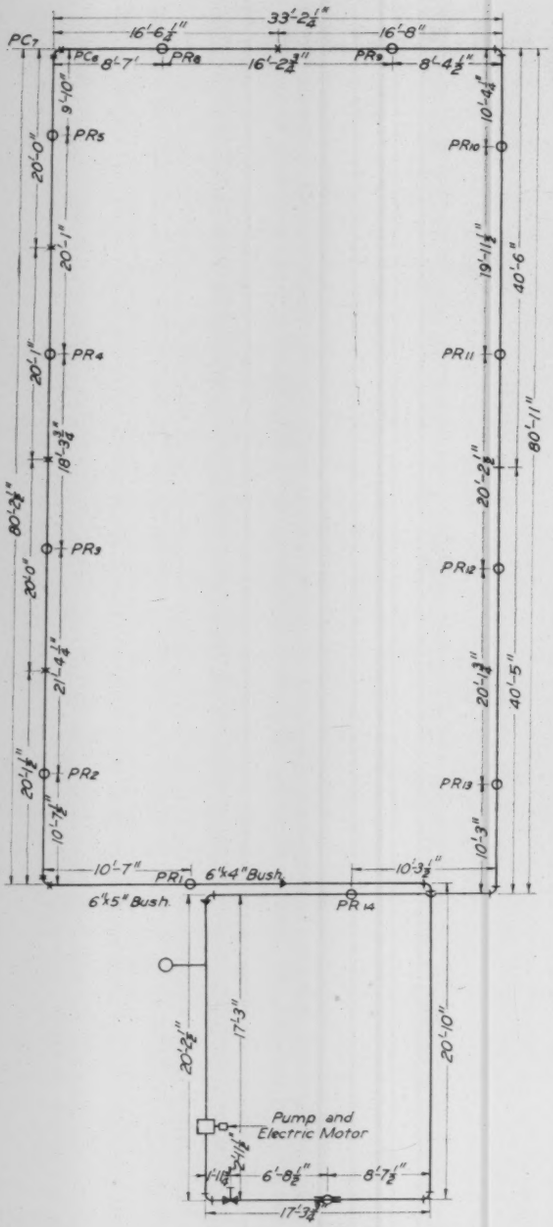


Fig. 1—Plan of 6-in. pipe set-up

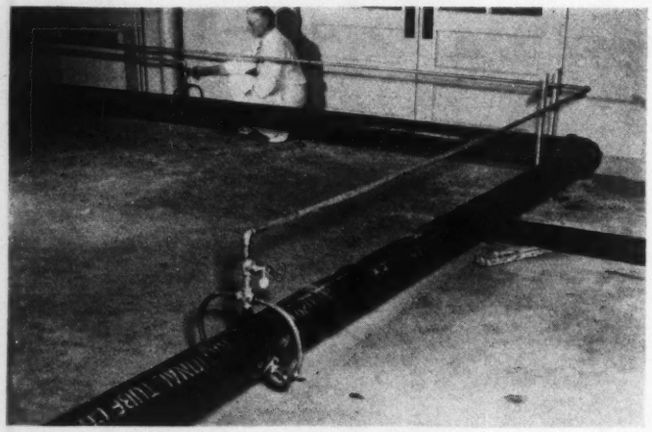


Fig. 3—Portion of 6-in. pipe with piezometer ring

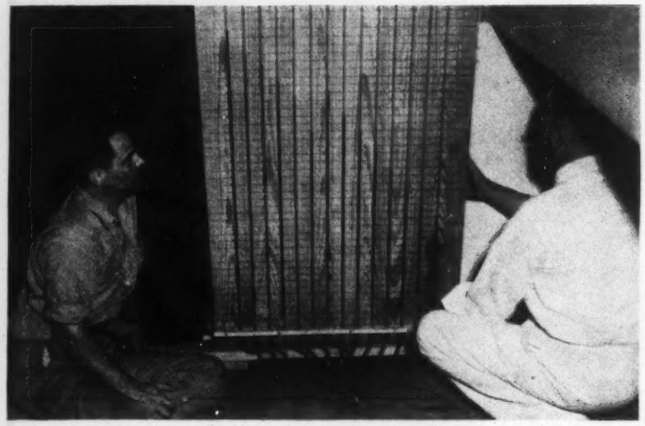


Fig. 4—Piezometer tubes

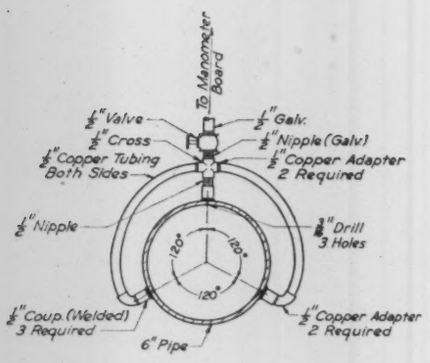


Fig. 2—Piezometer ring

4. Screwed 90-deg elbows of 6-in. pipe.
5. Welded 90-deg elbows of 6-in. pipe.

A secondary object was to determine the feasibility of using welded elbows to meter the flow of liquids in pipes.

Apparatus

The 6-in. pipe to be tested was supported on wooden blocks, near the floor of the laboratory of the Mechanical Engineering Department of the A & M College of Texas, as shown in Fig. 1. Fourteen piezometer stations were made in the pipe at the points marked PR1, PR2, etc., in Fig. 1. At each piezometer station were $\frac{3}{16}$-in. holes, spaced 120 deg apart, around the pipe. The holes were drilled and the inner surface of the pipe adjacent to the holes was scraped with a special scraping tool so that no burr was left at any of the holes. The three holes were connected by means of a piezometer ring, as shown in Figs. 2 and 3. The piezometer rings were connected, by means of carefully graded ½-in. pipe lines, to glass piezometer tubes mounted on a board, as shown in Figs. 4 and 5. The piezometer stations were located so that the losses of head for the prevailing velocity in the several typical portions of the pipe line were indicated by the differences in pressure at the corresponding piezometer stations. These losses of head were shown by the differences in the elevations of the water in the corresponding piezometer tubes. For example, some of the typical sections of the pipe line between piezometer stations are as follows:

PR1	to PR2	21 ft 2 in. of pipe and one welded elbow.
PR2	to PR3	21 ft 4¼ in. of pipe and one welded joint.
PR2	to PR5	59 ft 9 in. of pipe and two welded joints.
PR9	to PR10	18 ft 8¾ in. of pipe and one screwed elbow.
PR10	to PR11	19 ft 11½ in. of pipe.
PR10	to PR13	60 ft 3¾ in. of pipe and one screwed coupling.

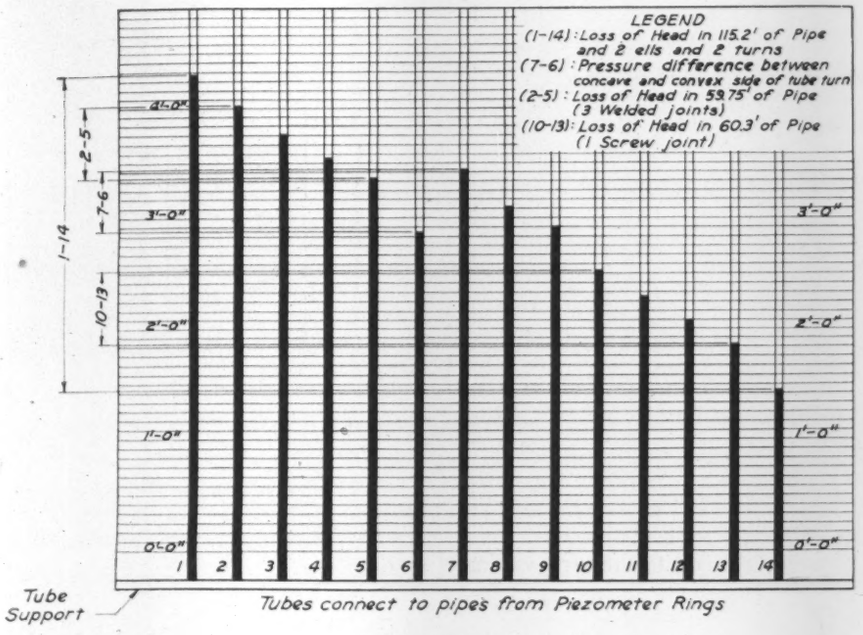

LEGEND
(1-14): Loss of Head in 115.2' of Pipe and 2 ells and 2 turns
(7-6): Pressure difference between concave and convex side of tube turn
(2-5): Loss of Head in 59.75' of Pipe (3 Welded joints)
(10-13): Loss of Head in 60.3' of Pipe (1 Screw joint)
1-14
2-5
7-6
10-13
4'-0"
3'-0"
2'-0"
1'-0"
0'-0"
1 2 3 4 5 6 7 8 9 10 11 12 13 14
Tube Support
Tubes connect to pipes from Piezometer Rings

Fig. 5—Piezometer tubes

Fig. 7—Orifice meter

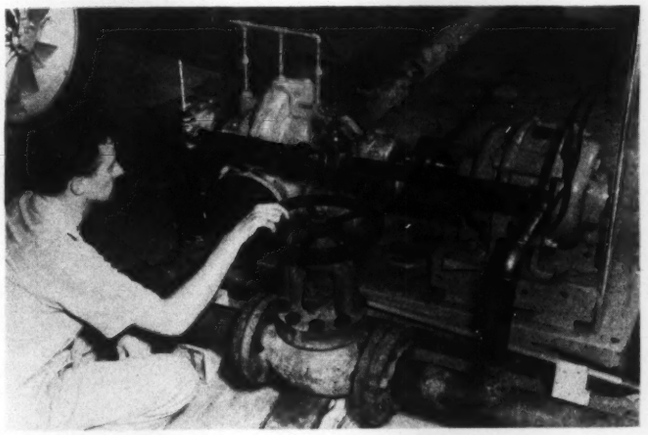


Fig. 6 (Left) — Motor, pump, and valve

Fig. 8 (Right) — Weighing tank in laboratory

Welded Joints

In the welded pipe used in these tests two of the welds were position welds and were made in three passes with $\frac{5}{32}$-in., $\frac{3}{16}$-in., and $\frac{5}{32}$-in. rods, used in the order named. All other welds were roll welds and were made with three beads as follows:

1. Stringer bead with a 5/32-in. rod.
2. Filler bead with a 3/16-in. rod.
3. Dresser bead with a 5/32-in. rod.

A $\frac{1}{16}$-in. space was left between pipe ends. The pipe ends had a 45-deg bevel, which made the included angle rather large, but satisfactory results were secured with the metallic arc welding.

It is evident that all welds were good because the test failed to indicate a measurable loss of head at any one of the welded joints.

Method of Testing

The water was circulated through the pipe line by means of a 1300 gpm pump direct-connected to a 15 hp motor, as shown in Fig. 6. The variations in the volume of water with temperature changes were accommodated by an open expansion tank. The quantity of water circulated through the pipe line was measured by means of an orifice meter (Fig. 7) consisting of a monel metal diaphragm with a 2.76-in. orifice and constructed according to the directions.[3]

The orifice meter was calibrated before the test was begun and again after the first seven series of tests had been completed. During the first calibration the water was pumped through the meter, in the same manner as during the 6-in. pipe test, and discharged into a weighing tank (Fig. 8). The time required to discharge 600 lb of water into a tank was determined by means of a stop-watch which was automatically started and stopped by the motion of the scale beam.

During the second calibration the water was allowed to flow through the meter from the college water mains and the time required to discharge 600 lb of water was determined by means of a stop-watch as before.

During both calibrations a buffer plate was suspended in the upper portion of the tank, so that the impact of the water on the tank and on the weighing platform was the same throughout the time during which

[3]See *Bulletin* No. 109, University of Illinois.

water was being discharged into the tank.

The calibration curve derived in this manner is shown in Fig. 9. The points shown by circles were determined during the first calibration, and those shown by triangles, during the second calibration. The curve, which was drawn before the

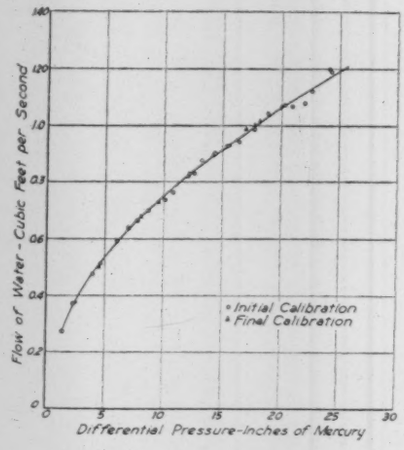


Fig. 9—Calibration curve

second calibration had been made, represented the points of the second calibration so well that no change was necessary after the second calibration.

In making a test run, the pump was started while the valve in the discharge line was closed (Fig. 6). The test was then begun either with the valve almost closed, so as to secure a low velocity (from 1 to 2 fps), or with the valve wide open, which gave a velocity of about 6 fps. The pipe system was purged of air by disconnection of the glass manometer tubes and allowing water to escape freely through the pipes leading to these tubes. After purging, the glass tubes were connected, and the water was allowed to circulate through the system until the water level in the manometer tubes became constant, at which time readings were taken at the flow meter and at the manometer tubes.

After the first set of readings had been taken, the valve setting was changed slightly to produce the next desired rate of flow. This process was repeated for different quantities of flow until the desired range of velocity had been attained in the 6-in. pipe. It required about three hours to complete such a run for each series of valve settings. Each series included about 14 sets of readings.

Nine series of tests were made. The data secured thereby were considered sufficient for the purpose of this investigation. The results of the fourth series are shown in Table 1, which is submitted here only as an illustration of the general method employed in recording and analyzing the data.

All calculations were made with the slide rule since it can be read as accurately as the orifice meter (Fig. 7) or the piezometer pressures (Fig. 4).

The values shown in Table 1 were transferred to Fig. 10 and lines were drawn through the resulting points to represent, as accurately as possible, the averages of the observed values. The lines of Fig. 10 show that the flow of the water in the 6-in. pipe was turbulent for velocities higher than 3 fps, because, for these higher velocities, the friction-heads are represented by straight lines when plotted to logarithmic coordinates against the corresponding velocities.

Table 2—Static Pressures
(Series 4 Inches of 70 F Water)

Piezometer Stations	Water Velocity—Fps			
	3	4	5	6
1	22.00	36.80	54.60	75.80
2	20.82	34.35	50.75	69.60
3	17.82	28.95	42.20	57.40
4	16.41	26.55	38.45	52.10
5	15.60	25.05	36.25	48.80
6	14.08	22.10	31.53	42.10
7	12.52	19.75	28.05	37.40
8	16.40	26.60	38.50	52.10
9	15.61	25.10	36.25	48.98
10	11.63	18.02	25.45	33.60
11	12.55	19.75	28.10	37.45
12	11.64	18.10	25.48	33.60
13	10.28	15.82	22.16	29.20
14	8.25	12.33	16.82	21.77

It is interesting to note that the results of this series of tests, as shown in Fig. 10, indicate that the transition from laminar to turbulent flow ended at a velocity of about 3 fps. The temperature of the water was about 90 F and Reynolds' number was approximately 194,000.

The equations of the straight portions of the lines were determined and recorded as shown in Fig. 10. With these equations, the piezometer pressures were calculated for velocities of 3, 4, 5, and 6 fps and recorded as shown in Table 2. From these values, shown in Table 2, the differences of the pressures at consecutive piezometer connections were calculated and recorded in Table 3.

The procedure described for the fourth series of tests was followed for the other seven series. The results of the first series of tests were discarded because it was believed that those of the later series were more accurate. The averages of the results secured for the remaining eight series were calculated and re-

Table 1—Observed Data in Test Four[a]

Orifice Meter in Hg	Pipe Water Temp. Deg F	Water Cu Ft per Sec	Tube Water Temp. Deg F	Static Pressure Readings—Inches of Water—for Piezometer Tubes (Numbered as Shown) 1	2	3	4	5	6	7	8	9	10	11	12	13	14	Water Velocity Ft-Sec
26.4	87.2	1.24	82.0	77.5	72.0	67.5	63.5	59.7	50.0	61.2	54.3	50.7	43.8	39.1	34.9	30.5	22.7	6.17
24.4	87.7	1.18	82.0	72.3	66.8	62.8	58.5	55.2	46.0	56.2	50.6	47.5	40.5	36.3	32.5	28.4	21.2	5.88
22.3	88.2	1.12	82.0	66.4	61.2	58.0	54.5	51.5	43.0	52.3	47.0	43.9	37.8	33.8	30.1	26.4	19.5	5.58
19.7	88.8	1.05	82.0	59.5	55.0	52.0	49.0	45.5	38.2	47.0	42.0	39.1	33.8	30.3	27.1	23.8	17.9	5.23
17.9	89.2	1.00	82.5	54.3	50.8	47.5	44.5	42.2	35.5	43.0	38.5	36.0	29.1	26.0	25.0	21.9	16.7	4.98
16.3	89.5	0.95	82.5	50.8	46.8	44.0	41.2	39.0	32.8	39.8	35.7	33.5	29.0	25.9	23.2	20.5	15.7	4.73
14.2	89.8	0.89	83.0	44.8	41.5	39.2	36.8	34.8	29.0	35.4	31.7	29.8	26.2	23.3	20.9	18.6	14.1	4.43
12.3	90.1	0.83	83.0	39.8	36.7	34.8	32.9	30.7	25.8	31.2	28.2	26.5	23.3	20.8	18.8	16.7	12.9	4.14
10.5	90.3	0.77	83.0	34.6	32.2	30.0	28.8	27.1	23.0	27.7	24.8	23.2	20.8	18.6	16.9	14.9	11.7	3.84
8.8	91.0	0.71	83.0	29.8	28.0	26.3	24.6	23.2	20.0	24.0	21.7	20.3	18.2	16.2	14.8	13.2	10.4	3.54
7.0	91.3	0.64	83.5	25.2	23.8	22.2	21.0	19.7	17.0	20.3	18.4	17.5	15.8	14.0	12.9	11.6	9.4	3.19
5.6	91.7	0.57	83.5	20.8	19.5	18.6	17.8	16.8	14.6	17.0	15.7	14.8	13.6	12.1	11.2	10.1	8.3	2.84
3.8	92.1	0.47	83.5	16.0	15.0	14.5	13.3	12.7	11.7	13.0	12.1	11.7	10.8	9.7	9.1	8.2	7.0	2.34
2.4	92.6	0.37	83.5	12.0	11.3	10.9	10.2	10.0	9.0	10.1	9.3	9.0	8.8	7.8	7.4	6.9	6.1	1.84
Total				603.8	560.6	528.3	496.6	468.1	395.6	478.2	420.7	403.5	351.5	313.9	284.8	251.7	193.6	
Average				43.1	40.0	37.7	35.5	33.45	28.2	34.2	30.05	28.8	25.1	22.4	20.35	17.95	13.8	

[a] Tables like this one, which shows the experimental results of Test 4, were prepared for the other nine tests but are not reproduced in this paper.

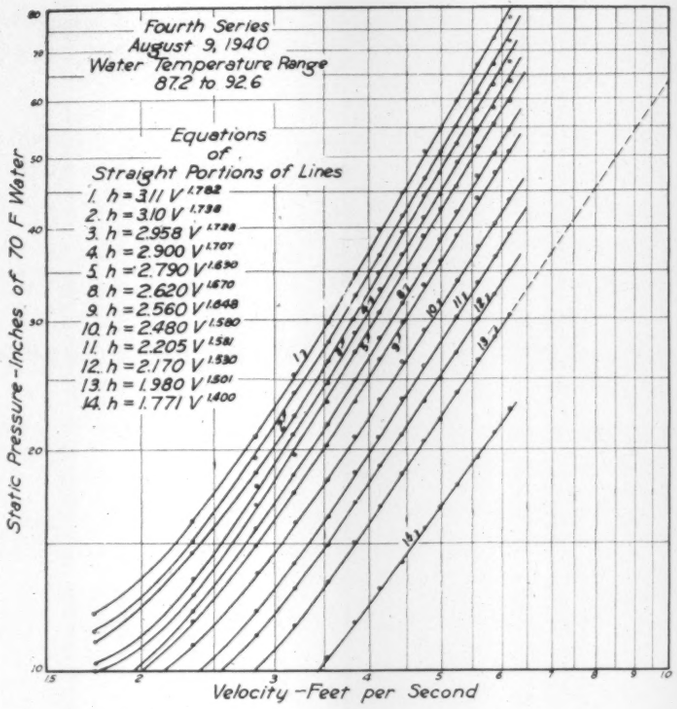

Fourth Series
August 9, 1940
Water Temperature Range
87.2 to 92.6
Equations
of
Straight Portions of Lines
1. h = 3.11 V 1.782
2. h = 3.10 V 1.738
3. h = 2.958 V 1.788
4. h = 2.900 V 1.707
5. h = 2.790 V 1.630
8. h = 2.620 V 1.670
9. h = 2.560 V 1.648
10. h = 2.480 V 1.580
11. h = 2.205 V 1.581
12. h = 2.170 V 1.530
13. h = 1.980 V 1.501
14. h = 1.771 V 1.400
Static Pressure - Inches of 70 F Water
Velocity - Feet per Second

Fig. 10—Static pressures of test 4

corded in Table 4 and represented graphically in Fig. 11, in which the center line of the pipe is taken as the coordinate axis and the static pressures above that at PR 14 are set off as ordinates at right angles thereto. The changes in pressure are represented by straight lines between consecutive piezometer connections except in the cases of the sections which include the elbows. In those cases, dotted lines were drawn to indicate, in a general way, the continuous change in static pressure, for the 6 fps velocity. In addition, the losses of head per foot of pipe, in milinches of water, are shown by figures on the respective lines for each of the seven sections and for velocities of 3, 4, 5 and 6 fps. It appears from Table 4 and from Fig. 11 that the total friction-head for the entire pipe line, consisting of 215.18 ft of 6-in. pipe, 2 welded elbows, 2 screwed elbows, 1 screwed coupling, and 4 welded joints, for velocities of 3, 4, 5, and 6 fps, were 14.70, 25.07, 37.86, and 52.66 in., respectively.

Having these data, the problem is to determine the friction-heads in 1 ft of pipe, 1 screwed elbow, 1 welded elbow, 1 screwed coupling, and 1 welded joint, for the several velocities.

When this study was planned it was assumed that:

1. That the piezometer connections were far enough from the adjacent elbows that the influence of the turbulence within the elbow would not extend beyond the nearest piezometer connection.

2. That the unit losses of head in Sections 10-11 and 12-13 would be the lowest of any in the pipe line and could be used as the friction-head in one foot of pipe, because those two sections contain no pipe fittings or joints.

3. That the unit loss of head in Section 11-12 would be slightly higher than those in the adjoining sections and that the difference could be used to calculate the friction-head of one screwed coupling.

4. That the unit losses of head in Sections 2-3, 3-4, and 4-5 would be slightly higher than those in Sections 10-11 and 12-13 and that the differences could be used to calculate the friction-head of one welded joint.

5. That the unit losses of head in Sections 1-2 and 5-8, would be higher than those in Sections 10-11 and 12-13, and that the difference could be used to calculate the friction-head of one welded elbow.

6. That the unit losses of head in Sections 9-10 and 13-14 would be higher than those in Sections 10-11 and 12-13 and that the difference could be used to calculate the friction-head of one screwed elbow.

The results of the experiment, recorded in Fig. 11, show that these assumptions were erroneous. The unit losses of head in Sections 10 - 11 and 12 - 13 are practically equal to each other but they are higher, instead of lower, than the unit loss of head in Section 11 - 12. The only apparent explanation of this condition is that the turbulence in Elbow 3 extended beyond PR 10 and that the turbulence in Elbow 4 extended beyond PR 13. It seems reasonable that the influence of Elbow 3 should have extended into its downstream tangent beyond PR 10, but it does not seem reasonable that the influence of Elbow 4 should have extended into its upstream tangent beyond PR 13. However, no other explanation is apparent at this time.

The result of this situation is that the unit losses of head in Sections 10 - 11 and 12 - 13 cannot be con-

Table 3 — Static Pressure Differences
(*Series* 4 *Inches of* 70 *F Water*)

Piezometer Stations	Water Velocity—Fps			
	3	4	5	6
1–14	13.75	24.47	37.78	54.03
1–2	1.18	2.45	3.85	6.20
2–3	1.09	1.94	3.20	4.70
3–4	0.81	1.60	2.55	3.30
4–5	1.02	1.81	2.85	4.26
5–8	1.41	2.40	3.75	5.30
8–9	0.81	1.50	2.20	3.30
9–10	1.52	2.95	4.72	6.70
10–11	1.55	2.45	3.42	4.65
11–12	0.89	1.73	2.60	3.80
12–13	1.35	2.20	3.29	4.40
13–14	2.03	3.49	5.34	7.43

Table 4 — Static Pressure Differences
(*Series* 2, 3, 4, 5, 6, 7, 8, 9
Inches of 70 *F Water*)

Piezometer Stations	Velocity—Fps				Section of Pipe Line
	3	4	5	6	
1–14	14.70	25.07	37.86	52.66	215.18 ft pipe + 2 tube turns + 2 elbows + 4 welds + 1 coupling
1–2	1.52	2.62	3.98	5.68	21.17 ft pipe + 1 tube turn
2–3	1.22	1.88	2.96	4.16	21.38 ft pipe + 1 weld
3–4	0.92	1.86	2.74	3.74	18.31 ft pipe + 1 weld
4–5	1.15	1.85	2.73	3.87	20.08 ft pipe + 1 weld
5–8	1.24	2.22	3.44	4.89	18.42 ft pipe + 1 tube turn
8–9	0.89	1.51	2.49	3.38	16.23 ft pipe + 1 weld
9–10	1.87	3.21	4.97	6.92	18.73 ft pipe + 1 elbow
10–11	1.31	2.21	3.18	4.42	19.96 ft pipe
11–12	1.12	1.89	2.78	3.85	20.21 ft pipe + 1 coupling
12–13	1.35	2.22	3.27	4.42	20.15 ft pipe
13–14	2.11	3.60	5.32	7.33	20.54 ft pipe + 1 elbow

sidered the unit friction-heads in 6-in. pipe.

In Sections 2 - 3, 3 - 4, and 4 - 5 the unit losses of head are very nearly alike; each section contains one welded joint and slight differences in the welded joints may account for the slight differences in unit head losses. It seems safe to conclude that the influence of the turbulences in Elbows 1 and 2 did not extend beyond the nearest piezometer connections. This seems reasonable because Elbows 1 and 2 are welded and have the same internal diameter as the pipe; whereas Elbows 3 and 4 are screwed elbows and have a larger internal diameter than the pipe and therefore create more turbulence than the welded elbows.

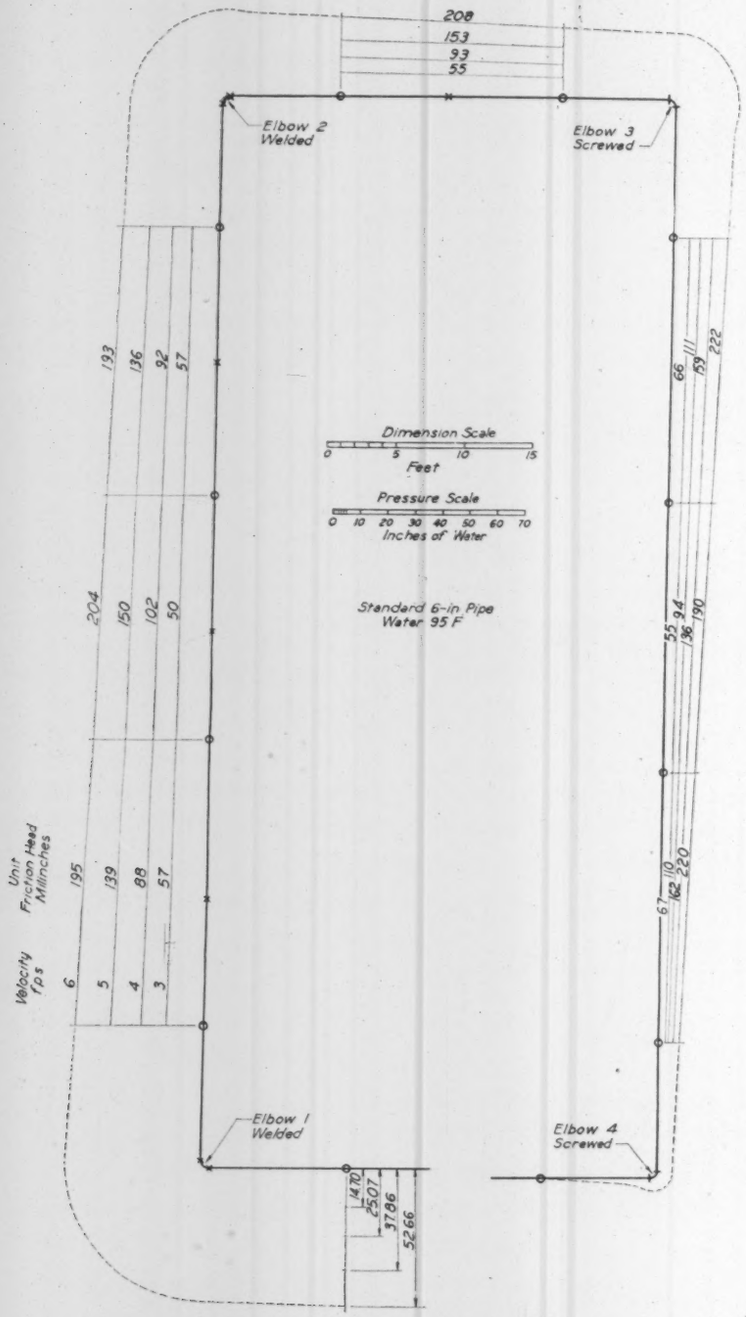


Fig. 11—Loss of head in 6-in. pipe and fittings

It seems permissible, therefore, to consider the unit losses of head in Sections 2-3, 3-4, and 4-5, as the friction-head in 1 ft of a 6-in. pipe line with a normal number of good welded joints. If the weighted mean values of these unit head losses are plotted on logarithmic paper, it will be found that the unit friction-head is

$$h = 0.00725\ v^{1.843} \quad \ldots\ldots\ldots\ldots\ldots\ldots (13)$$

where v is in feet per second and where h represents the friction-head in inches of 70 F water, per foot of pipe, when water at 95 F is flowing in the pipe.

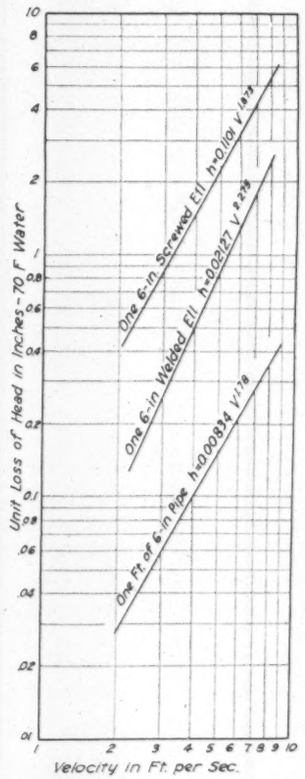

Unit Loss of Head in Inches - 70 F Water
Velocity in Ft. per Sec.
One 6-in Screwed Ell h=0.109 V 1.878
One 6-in Welded Ell h=0.02127 V 1.878
One Ft. of 6-in Pipe h=0.00834 V 1.78

Fig. 12—Unit loss of head in 6-in. pipe and elbows

The friction-heads in the two welded elbows can then be calculated by adding the losses of head in Sections 1-2 and those in Section 5-8 for velocities of 3, 4, 5, and 6 fps, and subtracting from the sum the losses of head calculated by Equation (13) for the straight pipe in those two sections, 39.63 ft, for

Tabe 5—Water Temperatures Fahrenheit At Beginnings and Ends of Tests

Series	Beginning	End
2	102.0	107.5
3	90.0	96.0
4	87.2	92.6
5	86.0	95.6
6	87.1	92.7
7	93.2	98.0
8	93.6	97.8
9	97.7	100.9
Average.....	92.1	97.6

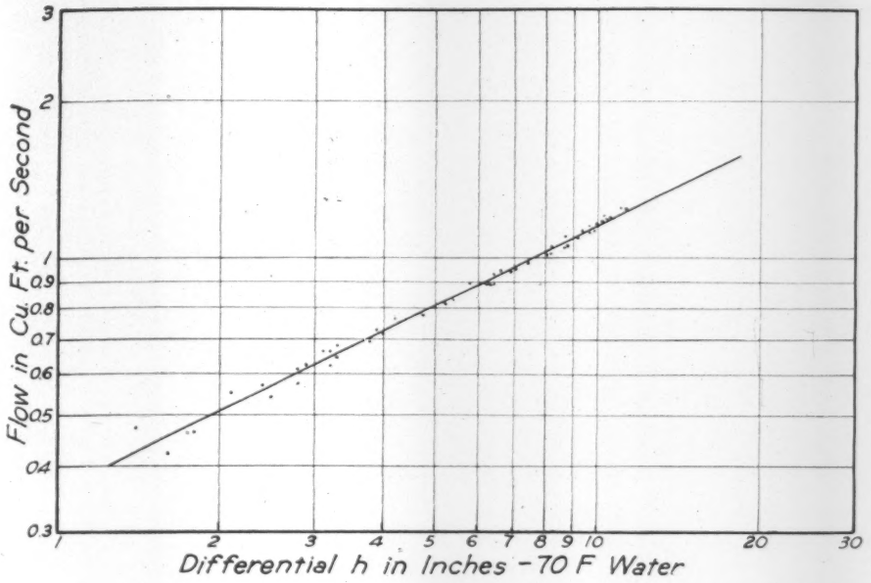


Fig. 13—Relation of flow and pressure difference at piezometer 6 and 7

the same velocities and plotting the results on logarithmic paper. This will show that the friction-head of one welded 6-in. elbow is

$$h = 0.023\ v^{2.3} \quad \ldots\ldots (14)$$

Instead of using only the losses of head found in Sections 2-3, 3-4, and 4-5 to determine the friction-head in 1 ft of a welded pipe line, as described previously, it may be better to use the losses of head in the entire straight portion of the experimental pipe line, *i.e.*, Sections 2-5, 8-9, and 10-13, as representing more nearly the type of pipe line ordinarily installed in practice.

If this is done, the unit friction-heads will be

$$h = 0.00834\ v^{1.78} \quad \ldots\ldots (15)$$

for one foot of 6-in. pipe;

$$h = 0.02127\ v^{2.275} \quad \ldots\ldots (16)$$

for one welded 6-in. elbow; and

$$h = 0.1101\ v^{1.873} \quad \ldots\ldots (17)$$

for one screwed 6-in. elbow, as shown in Fig. 12, where v is in feet per second and h represents the friction-head, in inches of 70 F water when water at 95 F is flowing in the pipe and fittings.

In the calculation to determine the friction-heads in screwed or welded elbows, it should be noted that the length of pipe connected to the elbow was measured from the piezometer station to the point of intersection of the center lines of the pipe. Consequently, the reported friction-head in one elbow is less than it would be if only the actual length of pipe, from the piezometer station to the elbow, had been used in the calculations. The foregoing method was adopted because it is believed to be common practice to measure pipe lines between the intersections of center lines of pipes, and because it simplifies friction-head calculations materially. This method reduces an elbow to a *no-length* fitting.[4]

To compare the results of Equation (15) with those of Equation (9) and with those shown by the chart in the GUIDE, Equation (15) should be modified so as to apply to 70 F water instead of to 95 F water. Assuming that, for this particular range of temperatures, the friction-head varies inversely as $F^{0.19}$, the modified form of Equation (15) will be

$$h = 0.0089\ v^{1.785} \quad \ldots\ldots (18)$$

The modified form of Equation (9) to apply to 6-in. pipe and to use inches instead of feet as a unit of measurement, is

$$h = 0.00826\ v^{1.965} \quad \ldots\ldots (19)$$

As an additional check on the experimental results, the losses of head were calculated according to the formula given by Nikuradse[5] for smooth pipe. This formula, with slight transformations, expressed in English units, and for values of R above 100,000 is

$$h = \left(0.0384 + \frac{2.652}{R^{0.237}}\right)\frac{L}{d}\,\frac{v^2}{2g} \text{ in.} \quad (20)$$

[4]The Friction of Water in Elbows, by F. E. Giesecke, C. P. Renning, and J. W. Knudson, Jr., (*Bulletin* No. 2712, University of Texas).
[5]*Forschungsheft*, by Nikuradse, 356, 1932.

The friction-heads, according to the three equations, and according to the chart in THE GUIDE 1940 are

	Velocities			
	3	4	5	6
Equation 19 (9).....	51	83	121	163
GUIDE, Equation 12.....	58	96	141	196
Equation 18 (15)	63	105	157	218
Nikuradse	46	77	115	160

According to Equation (20), the loss of head in 6-in. pipe, for velocities ranging from 3 to 6 fps, is about 12 per cent lower for 140 F water and about 20 per cent lower for 210 F water than for 70 F water.

In comparing the above values, it should be borne in mind that the values calculated by means of Equation (18) and (19) are for 70 F water, while those from THE GUIDE are for 140 F water and include an allowance for incrustation.

It is evident from this study that the charts in THE GUIDE should be revised, as soon as definite information regarding the influence of temperature and of incrustation from corrosion on the friction-heads is available.

Welded Elbow as a Meter

The possibility of using a welded elbow to measure the flow of liquids in pipes was checked. The differences between the static pressures at piezometers 6 and 7 were plotted in Fig. 13 against the volume of water flowing in the pipe, in cubic feet per second. The volume of water was determined from the readings of the orifice meter.

It appears from the graph that a welded elbow can be used as a meter; it also appears that the difference between the pressures on opposite sides of the elbow is an exponential function of the velocity, since the pressure difference plotted against volume is a straight line on logarithmic paper.

Conclusions

1. The friction-heads in couplings and in well-fabricated welds are very small and may be neglected in friction-head calculation.

2. The friction-head in one foot of a new and clean standard black 6-in. pipeline with a normal number of joints or couplings for water having a temperature of about 95 F is $h = 0.00834\ v^{1.78}$ in. of 70 F water, where v is in feet per second.

3. The transition period from

laminated flow to turbulent flow in 6-in. pipe seems to extend up to a velocity of about 3 fps. Additional studies should be made to determine the loss of head for velocities in the transition range.

4. The friction-head in one 6-in. screwed elbow is $h = 0.1101v^{1.873}$ in. of 70 F water, where v is in feet per second, when the temperature of the water flowing through the elbow is about 95 F and when the lengths of the connecting pipe lines are measured to the intersection of their axes. For the same conditions, the friction-head in one welded elbow is $h = 0.02127\ v^{2.275}$ in. of 70 F water.

5. A welded elbow can be used as a flow meter by measuring the difference in the static pressures between the concave and the convex sides and at the center of the elbow. This measurement is preferably made by means of an inclined manometer.

6. Additional research should be conducted to determine the effect of water temperature and also the effect of incrustation resulting from corrosion on the friction-head in pipes and fittings.

Acknowledgments

This investigation was sponsored by the American Society of Heating and Ventilating Engineers, and conducted at the Engineering Experiment Station, Texas A and M College, College Station, and acknowledgment is given for the assistance of the following: Bell and Gossett Co., A. M. Byers Co., Chicago Pump Co., Crane Co., Grinnell Co., Hoffman Specialty Co., Kewanee Boiler Corp., Minneapolis-Honeywell Regulator Co., National Tube Co., Nash Engineering Co., and Weil McLain Co. Acknowledgment is also given to Robert Cook, instructor in welding at A & M College of Texas, who did all the welding.

Heating, Ventilating, Air Conditioning, Guide 1941

In presenting the 19th edition of the Heating, Ventilating, Air Conditioning Guide for 1941 the Committee has attempted to provide the reader with recent authoritative and useful information. The 46 chapters have been rearranged and grouped into seven sections to make a more logical arrangement and to better correlate the Technical Data and the Catalog Data Sections.

Twenty-one chapters have been reviewed or rewritten for the new edition and many minor changes have been made in the other 25 chapters. An entirely new chapter on the Thermodynamics of Air and Water Mixtures is presented, based on the most recent information available on the subject. In addition to the Bulkeley Psychrometric Chart a new Mollier Diagram for Moist Air is included for use in analyzing air conditioning processes. The chapters on Cooling, Dehumidification and Dehydration, and Refrigeration are completely new and include the latest information available.

The chapter on Heat Transmission Coefficients and Tables contains some new data developed by recent research investigations. The table of climatic conditions in the chapter on Heating Load has been enlarged by adding a column on recommended design temperatures. In the chapter on Cooling Load will be found new data on solar heat transmission through walls, roofs and glass blocks, resulting from investi-

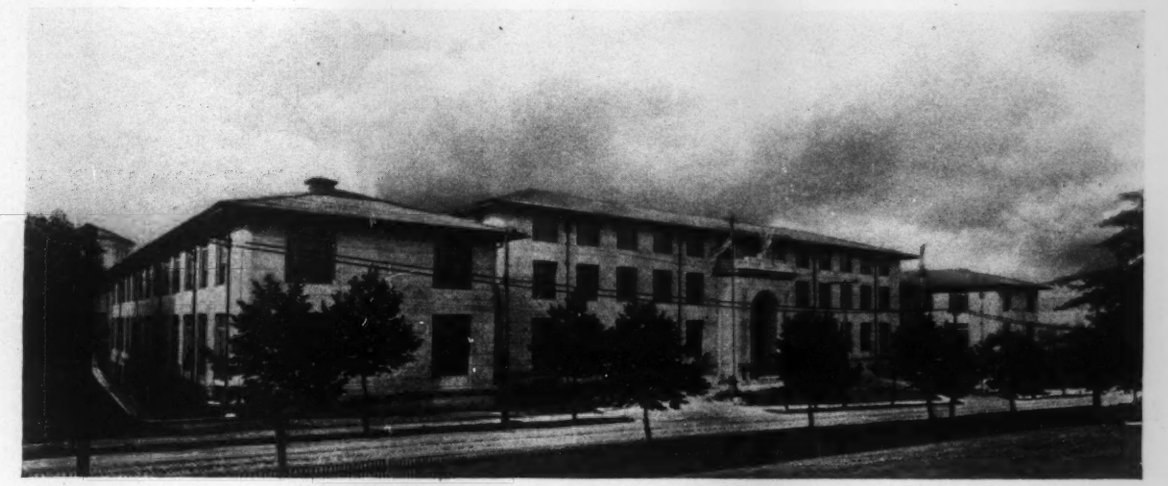
gations by the ASHVE Research Laboratory at Pittsburgh. Curves showing measured heat flow into structures replace the theoretical analysis frequently given in previous editions of The Guide.

New material on the design of cooling towers is included in the chapter on Spray Equipment. The Chapters on Air Duct Design, Sound Control and Automatic Control have been rewritten. The old chapter on Railway Air Conditioning has been rewritten and is now called Transportation Air Conditioning. Information on buses and automobiles has been added and references to air conditioning of ships and airplanes are included.

In the chapter on Terminology, some of the definitions have been revised to make them more exact and a table of specific heats has been added. Other chapters which have been reviewed and revised are as follows: Chimneys and Draft Calculations, Automatic Fuel Burning Equipment, Heat and Fuel Utilization, Central Systems for Comfort Air Conditioning, Fans, Air Distribution, Air Conditioning in the Treatment of Disease, Industrial Air Conditioning and Electric Heating.

An important part of The Guide is the Catalog Data Section, in which much useful information has been supplied by the various manufacturers, and this Section has been grouped into sub-divisions for convenience in locating data.

The new 1941 edition of The Guide comprises 46 chapters of technical data with 832 pages, the Catalog Data Section comprising 312 pages, the Roll of Membership of the Society and complete indices to the Technical and Catalog Data Sections. This handbook is again bound in a flexible blue cover with gold stamping, and is available at $5.00 per copy. A limited number of thumb-indexed copies will be available at $5.50 each for those who desire this particular feature.



Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is located

Radiation as a Factor in the Sensation of Warmth

By F. C. Houghten,* S. B. Gunst,** and J. Suciu, Jr.***
Pittsburgh, Pa.

SUMMARY—Tests are reported for two rooms oriented to give as nearly as possible the same exposure and heated by radiation and convection means. Reactions of subjects are reported giving sensations of draft, coolness and warmth for two types of heating. Appraisal of the radiation factor in influencing a person's feeling of warmth was studied with reference to relation between effective temperature and mean radiant temperature conditions

ONE of the most important fundamentals necessary for the application of comfort air conditioning is knowledge concerning the relation between the feeling of warmth of a person and his atmospheric environment. The atmospheric conditions affecting the feeling of warmth are the temperature of the air, its moisture content, and its movement. The relative importance of these factors is shown in the effective temperature scale, developed as one of the most important research investigations of the ASHVE Research Laboratory.

The fourth environmental factor influencing a person's feeling of warmth is radiation. The study of its significance was not included in the earlier research for the reason that it did not seem to be important in the American heating practice at that time, and it was, therefore, controlled by so insulating the rooms in which the studies were conducted, that the wall surfaces in view of the occupants were at approximately the same temperature as the air. With the growing adaptation of radiation to the American air conditioning practice, a technical advisory committee† was appointed during 1938 for the purpose of investigating some phases of this subject with the aid of a special grant of funds from the John B. Pierce Foundation. A comprehensive plan for carrying out the study was developed and a building consisting of two 12-ft x 15-ft x 8½-ft rooms connected by an observation room was built during 1939, in which an important phase of the investigations was carried out during the winter of 1939-40.

The results of the study to date indicate the degree to which the dry-bulb temperature, the moisture content, or the effective temperature of the environmental air must be decreased in order to compensate for the increased feeling of warmth resulting from a simultaneous increase in the radiation intensity, or vice versa.

Test Arrangements

A photograph of the test house located on the roof of the Bureau of Mines Warehouse Building is shown in Fig. 1. A floor plan of the test and observation rooms showing a number of the salient arrangements is given in Fig. 2. The shape and orientation of the building was designed to give as nearly as possible the same exposure to the walls of both rooms. As indicated in Fig. 2, the two rooms are identified as the East and West rooms. Characteristics of the construction in different parts of the test house are given in Table 1.

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†ASHVE Technical Advisory Committee on Radiation and Comfort, J. C. Fitts, *Chairman*, A. H. Barker, L. M. K. Boelter, R. E. Daly, E. R. Gurney, L. N. Hunter, A. P. Kratz, C. S. Leopold, D. W. Nelson, W. J. Olvany, G. W. Penney, W. R. Rhoton, C.-E. A. Winslow.

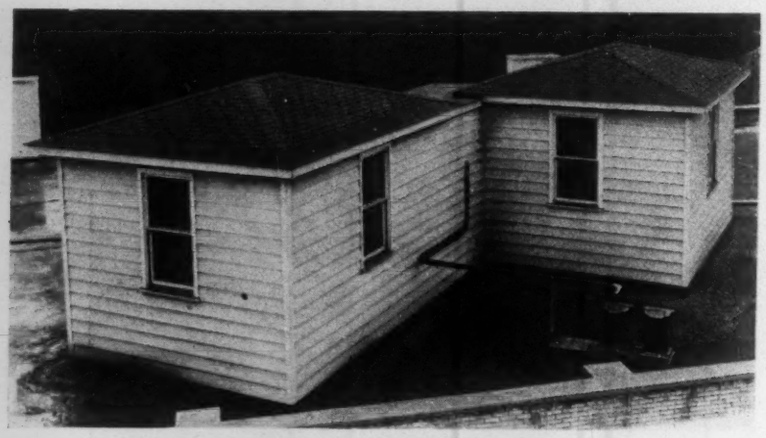

Fig. 1—View of radiation and comfort test building

In order to make the surroundings of the study as nearly typical of American heating practice as possible, the two rooms were so equipped that either might be heated by typical commercial hot water radiators or by a warm air circulation system. In Fig. 2 only one system of heating is shown in each room although during the testing program changes in the manner of heating for each room, that is, by radiation or convection, were made. In a given test the rooms are further identified as the radiation and convection rooms, sometimes abbreviated as *R* and *C*. Thus, in the arrangement in Fig. 2 is the west radiation room and the east convection room.

The heating requirements of both rooms were provided electrically so as to make available information concerning the variation in the heating load with outside weather conditions as well as in the two systems of heating. The radiation room was heated by two radiators of equal size, placed under the two windows, with the heat supplied by bayonet heaters screwed into both bottom connectors of each radiator. The electric heaters in each radiator were so arranged, with three-way switches, that one of the bayonet heaters could be supplied by hand control with any one of three constant rates of electrical input, while the heater in the other end of the radiator could be supplied with any one of three electrical input rates thermostatically controlled. This control provided a wide flexibility and, incidentally, a very uniform radiator surface temperature.

The convection room was equipped with an air circulating system installed in the space below the floor (convection room, Fig. 2). This system consisted of a fan exhausting air from a single floor grille (6 in. x 20 in.) placed below the window in the northwest side of the room and returning this air to the room through two 6 in. x 12 in. deflecting type grilles, placed 1¼ in. from the floor and 18 in. from each corner, in the southeast wall of the room. Thermostatic control of the two-speed fan gave an air delivery of 57 cfm through each supply grille or 115 cfm to the room until the delivered air reached 110 F. When the air supply temperature exceeded this value the fan was turned on high speed, delivering twice these volumes until the temperature dropped to 100 F when the fan was again returned to the low speed. This arrangement gave a maximum outlet temperature of 107 F with the fan running at a high speed on the coldest day on which a test was made. Heating units were located in two insulated sections of the duct near the warm air supply grilles. Each of these two heaters was made up of identical electrical heating units connected in series so that equal rates of heat input were supplied to each grille. A portion of this heat controlled by three rate switches was under hand control, and an independently controlled rate was under thermostatic control.

In order to maintain the same relative humidity in both rooms and in any one room throughout a given test period, water-evaporator humidifiers, electrically heated and under humidistatic control, were employed. One was placed near the return grille in the convection room

Table 1—Construction Characteristics of Test House

Surface	Construction	Transmission Coefficient (U)	Area in Each Room (Sq Ft)
Outside Walls*	1-in. yellow pine siding building paper 1-in. yellow pine sheathing 6-in. studding ½-in. rigid insulation board	0.19	378
Windows (2 in each room)	Single glazed, double hung, wood sash, weatherstripped	1.13	35
Floors**	1-in. yellow pine ½-in. rigid insulation board on 2x6 in. joists	0.27	176
Ceilings	½-in. rigid insulation board on 2x4 in. rafters	0.46	176
Roofs	Composition strip shingles 30-lb felt 1-in. yellow pine sheathing on 2x4 in. rafters	0.53	184
Roof and Ceiling		0.25	176

*Test rooms only.

**Since the upper and lower surfaces of the floor were maintained at the same temperature to within ± 1 F heat flow through the floor was limited to ± 0.27 Btu per square foot per hour.

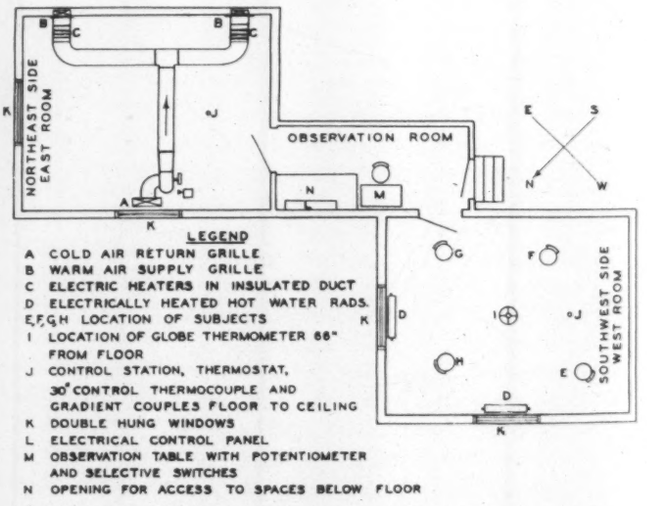
NORTHEAST SIDE EAST ROOM
OBSERVATION ROOM
SOUTHWEST SIDE WEST ROOM
LEGEND
A COLD AIR RETURN GRILLE
B WARM AIR SUPPLY GRILLE
C ELECTRIC HEATERS IN INSULATED DUCT
D ELECTRICALLY HEATED HOT WATER RADS.
E,F,G,H LOCATION OF SUBJECTS
I LOCATION OF GLOBE THERMOMETER 66" FROM FLOOR
J CONTROL STATION, THERMOSTAT, 30° CONTROL THERMOCOUPLE AND GRADIENT COUPLES FLOOR TO CEILING
K DOUBLE HUNG WINDOWS
L ELECTRICAL CONTROL PANEL
M OBSERVATION TABLE WITH POTENTIOMETER AND SELECTIVE SWITCHES
N OPENING FOR ACCESS TO SPACES BELOW FLOOR

Fig. 2—Floor plan of two 12 ft x 15 ft test rooms and observation room. Duplicate provisions in both rooms for radiators and air circulation systems, location of subjects, and measurement of all environmental conditions

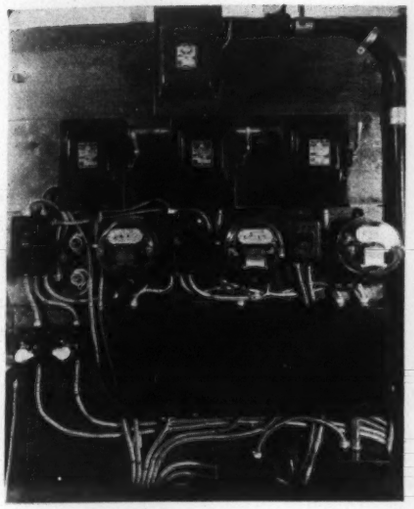

Fig. 3—View showing meters and switches used in radiation and comfort test building

and the other in front of a radiator in the radiation room so that the moisture was drawn into the recirculated air stream before it was heated. All electrical energy supplied in either room, including that supplied to the heaters, humidifiers, psychrometry fans, and lights passed through calibrated watt-hour meters which were read hourly during all tests. All electrical circuits were controlled from a switchboard, Fig. 3, in the observation room with the exception of the choice of heat input rates to each end of the two radiators.

The 30-in. basement space between the roof of the warehouse building and the floor of the test rooms was independently heated by steam radiators (located below the floor level in the observation room). The air flow over each of these was thermostatically controlled so as to give a zero ± 1 F temperature differential between the lower and upper surfaces of the floor.

A precision potentiometer with selective switches installed in the observation room made it possible to observe the temperature at 27 points throughout either room as indicated in Fig. 4, giving the temperature of the wall surfaces on the four sides of each room at seven points, the temperature of the air 6 in. away from five of these points, the temperature of the floor surface at one point, the temperature of the air 6 in. above this point, the temperature of the ceiling surface at two points, and the temperature of the air 24 in. below these two points. At the point indicated in Fig. 4 as the *control station* a ¼-in. pipe, flanged to the floor and ceiling, served to support the air thermocouples at the 6-, 30-, 78-, and 96-in. levels and the room thermostat at the 30-in. level.

Eight male college students between the ages of 19 and 22 years of age were selected as subjects for the study. These men were chosen as normal in every respect. They were given a thorough physical examination by a physician and only those whose metabolic rates, body temperatures, pulse rates, and other reactions which might have any thermal significance, were normal, were selected. These subjects wore normal attire for the winter season consisting of shorts; a sleeveless undershirt; shirt with collar and tie; socks; low cut shoes; and a medium weight, single-breasted suit including a vest, coat, and trousers. During a test they were seated in straight backed chairs and were allowed to engage in any sedentary activity which did not result in undue excitement or agitation, such as reading, preparing college lessons, or talking.

A globe thermometer for determining the mean radiant temperature (the integrated average temperature of all surroundings in view of a given point) was located in the center of each room at the 66-in. level. The sensitive element of a Pierce hot-wire anemometer was located in each test room near the globe thermometer. The globe thermometer temperatures and the anemometer observations were made from the observation room, thereby making it unnecessary for any observer to come near any of these instruments. The characteristics of the globe thermometer, the auxiliary instruments used therewith and the determination of the mean radiant temperature will be discussed below.

Test Operations

In the operation of any test on a given day the temperature of the rooms was kept near the desired level throughout the previous night and was then placed on the thermostatically controlled electrical heaters in the early morning. With the beginning of the preliminary test period at about 1:00 p.m., the eight male subjects were seated somewhat symmetrically in each room at points *E, F, G,* and *H,* Fig. 2, so that no parts of their bodies came within 2 ft of any wall surface or within 3 ft of any radiator surface. These four locations were further selected so that the subjects did not come within 3 ft of the control thermocouple, the thermostat or the entrance to the aspirating psychrometer. During the ¾-hour preliminary period, observations of the control temperature, a few of the wall surface temperatures, the wet- and dry-bulb temperatures and the globe thermometer temperatures were made.

At the close of the preliminary period the test proper was begun during which time the subjects remained in the room into which they originally entered for 80 min when they interchanged rooms for 80 min, and then again interchanged, so that

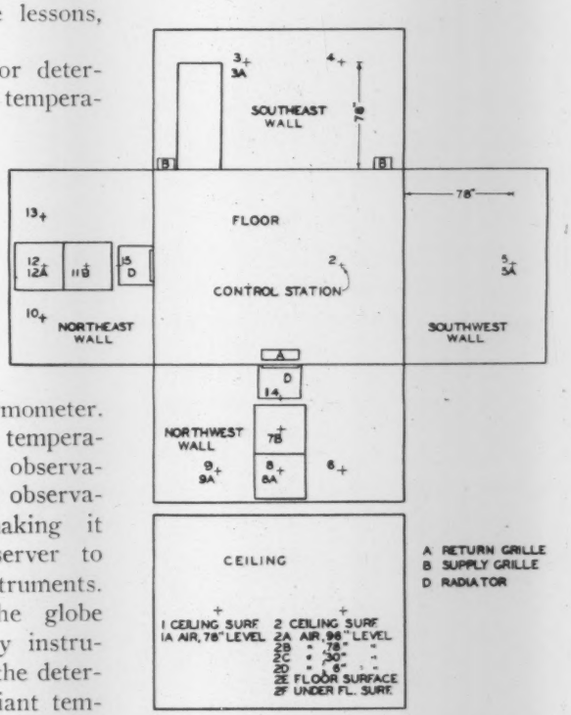

Fig. 4—Location of thermocouples in west room. Arrangement in east room identical with exception of door

(+) with single number indicates location of thermocouples on surfaces. Numbers and letters locate thermocouples in air 6 in. from surface unless otherwise indicated. Indicated distances are from floor level. Thermocouples No. 2 located on ¼-in. pipe flanged to floor and ceiling at control station

each group of subjects was again in its original room during the last 80 min of the 4-hour test period. During this period the subjects indicated their feelings of warmth and any local sensations of draft, coolness, and excessive warmth at 15-min intervals.

After the first few preliminary tests it was found desirable to maintain a constantly changing effective temperature in each room rather than a constant condition. Thus, the temperature was either allowed to rise or to fall in both rooms simultaneously at a constant rate during the four-hour period so that the subjects would slowly change from a condition where the average was slightly cool to a condition where the average was first comfortable and later slightly warm, or vice-versa. This was done in order to make available sufficient data so that, when plotted, they might indicate the particular time, and therefore the particular environmental conditions, at which the subjects were *ideally comfortable*.

During the 4-hour period frequent observations were made of all the temperatures, including the wet- and dry-bulb temperatures, globe thermometer temperature, both the exposed and the shielded air thermometer temperatures, and the air velocity (by the Pierce hot-wire anemometer).

Test Results

A log of one of the tests is given in Fig. 5 where the dry-bulb temperature, the wet-bulb temperature, and the resulting effective temperature at the 30-in. level *control station* are plotted for each room. An attempt was made to keep all three of these factors changing uniformly throughout the period of the test. Actually, the greatest attention was directed toward varying the effective temperature according to a predetermined curve and at the same time maintaining a constant relative humidity. This procedure necessitated the simultaneous control of the dry-bulb and the wet-bulb temperatures.

Also, in Fig. 5 is plotted for each room the mean radiant temperature as determined from the globe thermometer temperature, the globe temperature differential, the true air temperature, and the air velocity. A side wall temperature observed at a single point by thermocouple No. 5, Fig. 4, is also plotted.

In the lower part of the chart the average comfort votes for the four subjects, as observed at frequent intervals throughout the test, are plotted against the effective temperature of the conditions in each room at corresponding times. The comfort index here used was that employed in a number of earlier Laboratory studies during the past several years, in accordance with which a condition of *ideally comfortable* as far as the feeling of warmth is concerned is given an index number of *4*; the conditions of *comfortably cool* and *comfortably warm* are given index numbers of *3* and *5*, respectively; *cool* and *warm*, *2* and *6*, respectively; and *cold* and *hot*, *1* and *7*, respectively.

The two groups of subjects in each room, one during two 80-min periods, and the other during one 80-min period, are given separate symbols. It is noted that these give a very satisfactory plot through which a curve may be drawn whose intersection with the *4* ordinate gives the average effective temperature for comfort. The curve for the convection room is superimposed for comparison on the plot for the radiation room and indicates a lower index of warmth for any given effective temperature. Thus, in the radiation heated room on this day, the subjects averaged *ideally comfortable* at 68.8 deg ET, while in the convection heated room they indicated an average feeling of *ideally comfortable* at 70.0 deg ET. In the upper part of the chart there are plotted the electrical heat input to each room and the outside air temperature, ranging from 2.4 to 5.2 F, during the test period. This was the coldest day on which a test was made and required the highest rate of heat input. It will be observed that the radiation room required a somewhat greater rate of heat input.

After a preliminary investigating period of research into the best means of handling a test, 17 tests

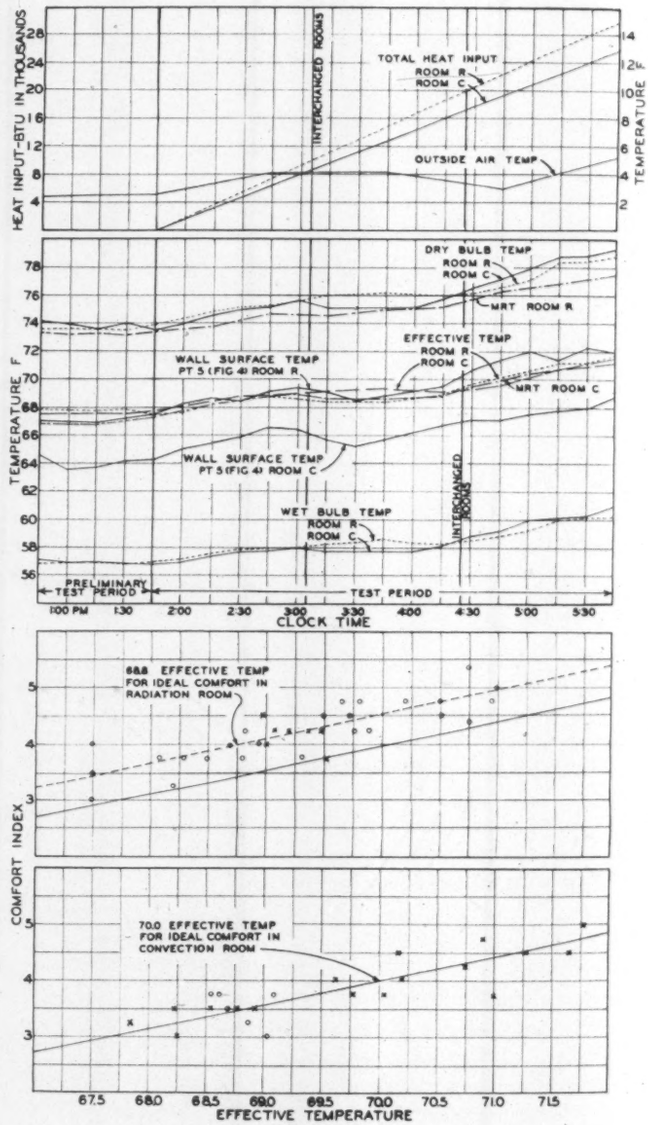

Fig. 5—Log of test No. 20 on January 19, 1940 (R—radiation room) (C—convection room)

Table 2—Data from 17 Tests in the Two Rooms

TEST NO.	DATE	OUTSIDE AIR TEMPERATURE	SOLAR RADIATION B.T.U./4 HRS.	AVERAGE OUTSIDE AIR VELOCITY DURING TEST MI./HR.	RADIATION ROOM	RADIATION ROOM: DRY BULB TEMP. 30" FROM FLOOR	RADIATION ROOM: DRY BULB TEMP. 78" FROM FLOOR	RADIATION ROOM: DRY BULB TEMP. AVERAGE (30" AND 78")	RADIATION ROOM: WET BULB TEMPERATURE	RADIATION ROOM: EFFECTIVE TEMPERATURE	RADIATION ROOM: DEGREES E.T. CHANGE FOR 1 COMFORT INDEX CHANGE	RADIATION ROOM: MEAN RADIANT TEMP.	RADIATION ROOM: AVERAGE DB - MRT	RADIATION ROOM: AIR VELOCITY FT/MIN.	RADIATION ROOM: CEILING - FLOOR AIR TEMPERATURE DIFFERENCE	RADIATION ROOM: HEAT INPUT B.T.U./HR.	RADIATION ROOM: HEAT INPUT B.T.U./HR./°F DIFFERENCE INSIDE - OUTSIDE	CONVECTION ROOM: DRY BULB TEMP. 30" FROM FLOOR	CONVECTION ROOM: DRY BULB TEMP. 78" FROM FLOOR	CONVECTION ROOM: DRY BULB TEMP. AVERAGE (30" AND 78")	CONVECTION ROOM: WET BULB TEMPERATURE	CONVECTION ROOM: EFFECTIVE TEMPERATURE	CONVECTION ROOM: DEGREES E.T. CHANGE FOR 1 COMFORT INDEX CHANGE	CONVECTION ROOM: MEAN RADIANT TEMP.	CONVECTION ROOM: AVERAGE DB - MRT	CONVECTION ROOM: AIR VELOCITY FT/MIN.	CONVECTION ROOM: CEILING - FLOOR AIR TEMPERATURE DIFFERENCE	CONVECTION ROOM: HEAT INPUT B.T.U./HR.	CONVECTION ROOM: HEAT INPUT B.T.U./HR./°F DIFFERENCE INSIDE - OUTSIDE	HEAT INPUT ROOM R IN % ROOM C: % B.T.U./HR.	HEAT INPUT ROOM R IN % ROOM C: % - B.T.U./HR./°F. DIFFERENCE INSIDE - OUT.
A	B	C	D	E	F	G_R	H_R	T_R	J_R	K_R	L_R	M_R	N_R	O_R	P_R	Q_R	R_R	G_C	H_C	I_C	J_C	K_C	L_C	M_C	N_C	O_C	P_C	Q_C	R_C	S	T
14	1/11	36.3	0	9	W	73.3	74.3	73.8	60.1	68.5	2.0	72.3	1.5	30	2.1	1728.8	46.1	74.3	74.9	74.6	60.5	69.1	2.0	71.2	3.4	31	-0.1	1471.1	38.4	118	120
15	1/12	39.9	0	7	W	73.0	73.3	73.1	60.7	68.5	2.2	71.6	1.5	35	1.7	1746.0	52.6	73.4	73.4	73.4	61.0	68.7	2.2	71.4	2.0	31	0.1	1507.2	45.0	116	130
16	1/15	27.9	25	19	W	74.4	76.4	75.4	58.7	68.5	2.5	74.0	1.4	25	2.7	3974.3	83.7	75.0	76.5	75.8	59.7	69.0	2.3	71.7	4.1	31	1.7	3312.3	69.2	120	121
17	1/16	20.5	0	16	W	74.2	76.2	75.2	57.8	68.2	2.5	72.7	2.5	25	4.9	4781.9	87.4	76.2	78.2	77.2	57.8	69.2	2.5	72.9	4.3	30	2.2	4453.8	78.6	107	111
18	1/17	17.2	75	9	W	75.1	76.6	75.8	57.6	68.6	3.7	74.6	1.2	21	2.3	2341.7	39.9	77.3	78.8	78.1	58.1	69.9	3.4	73.3	4.8	25	2.0	2973.7	48.8	79	83
19	1/19	13.0	400	14	W	75.0	75.7	75.3	57.2	68.3	3.1	74.4	0.9	20	3.3	3914.9	62.8	78.0	79.4	78.7	58.7	70.3	3.3	74.0	4.7	45	2.5	5026.1	76.5	78	82
20	1/19	4.7	0	14	W	75.1	77.1	76.1	59.0	68.8	2.3	74.4	1.7	23	3.5	7796.9	109.2	77.0	78.0	77.5	60.5	70.0	2.3	70.3	7.2	32	2.2	6821.4	93.7	114	118
22	1/23	26.5	420	9	W	75.2	77.1	76.1	58.1	68.9	3.6	73.2	0.9	27	2.6	4590.5	92.6	76.3	76.7	76.5	59.0	69.6	4.1	71.0	5.5	40	1.7	3551.5	71.0	129	130
23	1/24	27.5	0	14	W	74.8	76.4	75.6	58.1	68.6	2.4	73.2	2.4	25	3.0	5368.5	115.8	75.3	75.0	75.1	59.1	69.1	2.2	70.0	5.1	40	3.0	4047.4	85.1	138	136
24	2/1	33.9	69	12	E	75.1	77.6	76.3	58.2	68.7	2.3	74.0	2.3	22	3.0	4908.6	115.8	76.6	78.6	77.6	60.0	69.9	2.5	74.1	3.5	45	3.0	4597.2	105.2	107	110
25	2/2	17.0	457	16	E	75.0	76.4	75.7	58.0	68.5	2.3	74.5	1.2	25	0.9	6094.2	103.8	76.0	78.9	77.4	58.3	69.3	2.5	74.0	3.4	32	2.3	5919.8	98.0	103	106
26	2/3	38.6	42	5	E	73.2	75.7	74.4	60.9	68.5	1.9	72.2	2.2	24	3.0	3472.2	97.0	73.3	73.9	73.6	60.5	68.3	3.7	70.6	3.0	35	2.1	3179.4	90.8	109	107
27	2/7	32.0	0	10	E	72.7	73.7	73.2	60.0	68.0	3.4	69.4	3.3	46	3.2	4544.4	110.3	73.6	74.3	73.9	61.0	68.8	2.9	71.2	2.7	30	1.8	4206.8	100.4	108	110
28	2/8	40.2	309	6	E	73.5	73.4	73.4	60.7	68.5	2.8	72.2	1.2	25	0.5	1666.6	50.2	72.8	72.2	72.5	61.6	68.8	1.7	71.4	1.1	26	0.8	2161.8	66.9	77	75
30	2/13	32.3	0	6	E	74.0	74.2	74.1	61.6	69.1	3.5	70.4	3.7	25	3.1	4767.7	114.1	75.2	74.9	75.0	62.9	70.1	3.5	71.6	3.4	30	1.2	3689.3	86.4	129	132
31	2/14	26.6	197	23	E	74.6	74.2	74.4	59.2	68.7	2.0	72.0	2.4	25	1.3	3134.2	65.6	75.6	75.6	75.6	62.6	69.7	2.5	72.2	3.4	41	1.2	2964.5	60.5	106	109
32	2/15	30.8	1106	13	E	75.0	75.1	75.0	59.3	69.0	2.7	73.5	1.5	26	0.2	1302.6	29.5	75.5	75.5	75.3	60.2	69.5	2.9	73.1	2.2	32	1.1	1580.6	35.5	83	82
AVERAGE VALUES						74.3	75.5	74.9	59.2	68.6	2.7	73.0	1.9	27	2.4	3902.0	80.9	75.4	76.2	75.8	60.1	69.4	2.7	72.0	3.8	33.8	1.7	3615.5	73.5	108	110

*Windows in radiation room blanked off with 1 in. structural board insulation.

were conducted and the results plotted similarly to Test No. 20 in Fig. 5. Important results for these tests are given in Table 2, together with certain averages in the bottom line for all tests. Temperatures observed throughout the two rooms at times during the different tests when the subjects averaged *ideally comfortable* are given in Table 3. The locations of the observation points are indicated in Fig. 4. The distances, expressed in inches, for the different points are measured from the floor; therefore, zero represents the thermocouple attached to the floor surface; 102 in., the thermocouple attached to the ceiling surface.

Analyses of Test Results

A comparison of the test results from the radiation and convection rooms indicates a universally lower effective temperature for comfort in the radiation room than for the convection room. This is accounted for by the warming effect on the subjects by direct radiation from the radiators and that reflected and reradiated from other surfaces within the room. The average values in Table 2 indicate that the average temperature for comfort for all tests in the radiation room was 68.6 deg ET, while for the convection room the effective temperature had to be an average of 0.8 deg higher, or 69.4 deg ET. This is better shown in Figs. 6 and 7, where the outside air temperature at the time when the subjects were *ideally comfortable* within the room is plotted against the effective temperature required for ideal comfort. Tests were conducted when outside temperatures ranged from 4 F to 40 F.

The points *X*, fixing the upper ends of the curves at 73.6 F outside temperature, were determined independently in the psychrometric rooms of the Laboratory for the purpose of this study. This pro-

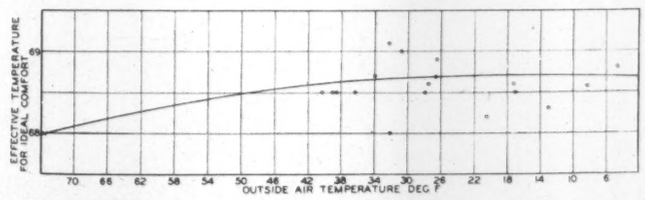

Fig. 6—Relation between effective temperature at which comfort was realized and concurrent outside air temperature for radiation room

(×) dry-bulb at 40 per cent RH and 68 deg ET where subjects were comfortable with no difference between air and mean radiant temperatures

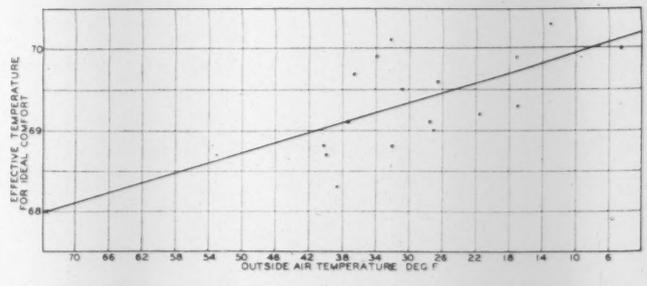

Fig. 7—Relation between effective temperature at which comfort was realized and concurrent outside air temperature for convection room

(×) dry-bulb at 40 per cent RH and 68 deg ET where subjects were comfortable with no difference between air and mean radiant temperatures

Table 3—Temperatures Observed Throughout the Two Rooms at a Time During the Different Tests When the Subjects Averaged Comfortable

		Test Number	Date	Outside Air	Time of Observation	Ceiling			Central Control					Southeast Wall			Southwest Wall		Northwest Wall						Northeast Wall					
						1 102″ X	1A 78″ Z	2 102″ X	2A 96″ Y	2B 78″ Z	2C 30″	2D 6″ Y	2E 0″ X	3 78″ X	3A 78″ Y	4 78″ X	5 78″ X	5A 78″ Y	6 78″ X	7 48″ G	8 78″ G	8A 78″ YG	9 78″ X	9A 78″ Y	14 24″ R	10 78″ X	11 48″ G	12 78″ G	12A 78″ YG	13 78″ X	15 24″ R
Room R	West Room	14	1–11	36.3	3:30	69.5	74.3	68.2	74.3	74.3	73.3	73.2	72.6	70.1	74.1	68.5	68.7	73.8	69.0	55.6	55.8	74.7	69.5	74.3	96.4	68.7	58.5	56.6	75.0	69.3	96.7
		15	1–12	39.9	2:45	69.1	73.4	68.3	74.1	73.3	73.0	72.4	71.8	69.4	73.3	69.3	68.5	73.2	69.1	59.4	57.7	73.9	69.2	73.4	99.4	68.8	58.5	57.0	74.1	69.0	100.1
		16	1–15	27.9	3:30	70.8	76.3	68.9	76.4	76.4	74.4	73.7	72.0	69.9	76.3	69.7	68.9	75.0	69.7	49.9	48.4	78.6	69.1	75.9	120.2	68.6	51.9	50.5	78.6	69.2	121.7
		17	1–16	20.5	2:09	70.2	76.4	67.9	77.6	76.2	74.2	72.7	70.0	69.3	76.9	67.1	67.5	76.4	67.2	47.7	45.8	83.0	68.2	77.4	131.3	67.0	47.7	50.1	83.1	67.7	133.0
		18	1–17	17.2	4:33	71.9	76.7	70.3	77.4	76.6	75.1	75.1	75.5	71.8	76.7	69.7	70.1	76.6	71.2	Window Blanked Off			69.9	76.3	93.8	68.9	Window Blanked Off			73.3	85.1
		19	1–18	13.0	5:30	69.7	77.6	71.1	77.0	75.7	75.0	73.7	73.5	70.4	77.2	69.0	68.5	76.1	69.4				68.0	76.2	127.1	68.0				71.7	93.4
		20	1–19	4.7	3:15	70.0	78.1	69.2	77.6	77.1	75.1	74.1	72.4	70.4	77.6	68.6	67.8	77.1	67.4	39.4	37.6	79.3	67.6	77.2	137.2	65.9	38.5	41.3	79.5	66.2	136.6
		22	1–23	26.5	2:45	73.6	78.7	70.3	78.6	77.1	75.2	75.6	75.3	73.1	78.2	70.5	71.5	78.1	70.6	59.0	56.2	83.3	70.3	78.6	124.7	70.0	56.0	54.9	84.0	70.6	122.0
		23	1–24	27.5	3:48	71.1	76.9	69.5	76.7	76.4	74.8	73.7	70.6	69.8	78.3	70.8	69.3	77.0	71.0	51.8	50.8	82.6	71.7	78.6	127.5	68.5	52.0	51.8	82.6	72.2	129.7
	East Room	24	2–1	33.9	3:15	70.5	75.4	72.4	77.0	77.6	75.1	74.0	74.1	69.9	74.9	71.1	69.6	76.4	70.3	57.6	55.8	85.6	69.4	74.8	147.0	71.8	50.8	52.7	74.5	69.7	81.8
		25	2–2	17.0	3:45	70.7	76.1	70.4	74.8	76.4	75.0	74.9	74.1	68.7	75.8	70.9	71.1	74.9	68.3	45.1	44.4	79.6	69.0	76.1	128.7	67.6	49.8	48.9	78.8	67.9	126.6
		26	2–5	38.6	4:30	68.6	75.4	70.9	75.8	75.7	73.2	72.5	72.0	69.0	74.7	68.2	68.3	73.8	69.0	56.8	55.6	77.5	69.5	74.9	118.4	68.3	59.8	59.8	78.5	68.5	115.1
		27	2–7	32.0	3:18	67.7	73.6	68.3	75.5	73.7	72.7	72.3	72.0	66.5	74.0	67.9	66.7	73.1	71.3	53.1	50.7	75.9	67.5	74.6	111.3	66.9	54.4	53.9	76.0	67.3	109.8
		28	2–8	40.2	4:15	70.5	74.3	70.1	73.6	73.4	73.5	73.1	73.0	68.6	73.4	69.2	69.9	73.5	68.8	55.6	54.8	72.5	68.3	73.6	95.1	68.7	54.3	54.7	73.6	69.0	94.1
		30	2–13	32.3	4:13	68.3	74.4	69.8	76.2	74.2	74.0	73.1	72.6	67.7	74.8	68.7	69.3	74.0	71.0	56.4	55.5	77.3	68.2	74.5	118.0	67.0	54.5	55.5	77.4	67.0	117.4
		31	2–14	26.6	4:26	69.6	75.8	70.6	75.2	74.2	74.6	74.2	73.6	68.8	75.3	70.8	69.4	74.7	72.0	51.1	48.8	76.3	69.3	75.3	112.3	68.2	51.1	50.0	77.5	68.4	110.8
		32	2–15	30.8	3:45	69.5	74.7	69.4	75.1	75.1	75.0	74.9	74.8	71.6	75.1	71.6	73.6	75.0	69.4	50.4	50.2	74.4	69.1	74.7	97.6	68.4	51.5	52.7	74.1	68.4	99.2
Room C	East Room	14	1–11	36.3	3:30	69.5	74.6	70.9	74.6	74.9	74.3	74.7	77.0	69.7	74.7	70.0	69.8	74.2	71.0	57.9	58.1	73.0	69.6	74.2	.....	69.4	56.0	56.4	72.6	69.4	.....
		15	1–12	39.9	2:30	69.2	73.3	69.3	74.3	73.4	73.4	74.2	74.5	69.5	73.5	69.8	69.3	74.5	70.4	56.6	56.2	73.2	70.3	73.9	.....	69.3	56.5	56.5	73.1	69.5	.....
		16	1–15	27.9	3:30	71.7	74.2	72.0	75.9	76.5	75.0	74.2	75.7	68.5	74.8	70.0	70.0	75.8	69.0	45.6	48.2	73.4	67.6	73.9	.....	67.5	46.4	47.6	72.6	68.4	.....
		17	1–16	20.5	4:00	70.6	79.6	72.8	78.4	78.2	76.2	76.2	77.3	70.1	77.0	71.7	71.3	77.3	71.2	53.3	55.4	76.7	70.6	79.2	.....	70.3	48.6	49.1	77.1	70.4	.....
		18	1–17	17.2	4:30	70.9	77.7	71.2	78.7	78.8	77.3	76.7	76.5	68.6	76.9	69.9	71.5	77.7	70.4	45.7	50.8	77.1	69.8	79.3	.....	67.7	47.0	47.5	75.8	68.4	.....
		19	1–18	13.0	3:15	70.1	80.3	73.7	81.2	79.4	78.0	78.7	77.8	70.8	80.7	73.9	72.1	80.1	72.3	42.9	43.3	77.4	70.9	79.9	.....	70.0	44.8	46.4	78.0	69.9	.....
		20	1–19	4.7	2:24	69.6	77.6	70.0	78.0	78.0	77.0	75.8	74.2	67.0	77.5	69.4	67.9	77.3	69.8	33.6	35.8	76.1	66.4	74.3	.....	65.1	38.6	40.7	74.6	65.9	.....
		22	1–23	26.5	3:15	68.2	75.2	71.4	75.7	76.7	76.3	75.3	73.8	68.7	74.8	70.2	70.0	76.8	69.5	49.4	50.1	74.2	68.8	74.9	.....	64.8	46.6	49.4	75.0	63.8	.....
		23	1–24	27.5	3:05	69.7	74.8	70.7	75.9	75.0	75.3	72.8	70.7	65.7	74.3	67.2	67.5	75.7	67.1	45.1	45.8	75.4	66.7	75.4	.....	65.9	46.1	48.7	73.4	66.2	.....
	West Room	24	2–1	33.9	2:45	72.5	78.4	72.2	78.2	78.6	76.6	73.7	71.4	71.7	77.2	71.0	70.8	77.9	72.5	50.4	51.1	76.3	70.6	77.9	.....	69.7	51.8	50.2	77.2	76.4	.....
		25	2–2	17.0	4:03	71.8	79.2	72.5	79.8	78.9	76.0	73.6	72.0	72.9	80.3	71.2	71.4	78.8	68.3	42.3	42.4	78.0	69.8	78.6	.....	69.2	42.6	42.8	78.1	72.6	.....
		26	2–5	38.6	3:11	69.5	73.8	69.3	74.4	73.9	73.3	71.3	70.6	69.6	74.8	68.0	69.2	74.3	68.6	50.6	51.6	73.2	68.5	73.6	.....	67.8	53.1	53.7	73.3	70.7	.....
		27	2–7	32.0	3:50	69.1	75.0	69.5	75.2	74.3	73.6	71.9	70.8	69.1	75.5	68.4	68.6	73.9	67.5	47.8	48.1	74.2	67.8	73.6	.....	67.2	49.7	51.6	74.5	71.2	.....
		28	2–8	40.2	4:51	68.3	71.6	70.5	71.9	72.2	72.8	72.1	72.2	69.3	71.9	68.3	69.8	72.6	67.9	54.5	54.4	72.7	68.2	72.6	.....	67.4	54.5	55.4	70.9	68.0	.....
		30	2–13	32.3	4:45	68.1	75.1	69.1	76.0	74.9	75.2	74.0	71.4	68.8	74.6	67.7	69.2	74.9	71.4	51.4	49.7	72.4	67.2	71.1	.....	67.7	49.0	49.8	74.3	72.3	.....
		31	2–14	26.6	3:24	69.4	75.8	71.5	72.9	75.6	75.6	74.5	73.9	70.1	76.5	69.4	71.5	76.5	69.5	50.0	51.9	75.8	70.7	76.0	.....	68.8	48.8	50.9	75.6	73.8	.....
		32	2–15	30.8	3:40	70.9	75.2	69.8	75.6	75.5	75.5	74.0	73.9	72.6	74.7	71.3	73.4	76.0	69.6	49.5	50.3	73.0	68.8	73.5	.....	68.3	50.2	50.3	73.2	62.8	.....

Location of observation points as indicated in Fig. 4. Distances measured from floor,

Therefore:

0 in. = floor surface
102 in. = ceiling surface
X = surface temperatures
Y = air temperature 6 in. from surfaces
Z = air temperature 24 in. from surfaces
G = Glass surface temperature
R = radiator surface temperature

cedure will be discussed later. It represents an equilibrium condition where the outside dry-bulb, the inside dry-bulb, and the wall surface temperatures were all the same with 68 deg ET and 40 per cent relative humidity.

While there is considerable scattering of the points, these tests indicate definite trends in the results as shown by the curves as drawn. For the radiation room there is very little variation in the effective temperature for comfort given by the test results, while for the convection room there is a definite trend, indicating a desire for a higher effective temperature with a drop in the outside temperature. This may logically be accounted for by the fact that with decreasing outside temperature the radiators had to be maintained at higher temperatures with increasing radiant effects within the room. It appears that in the radiation room this effect just about compensates for the lowering

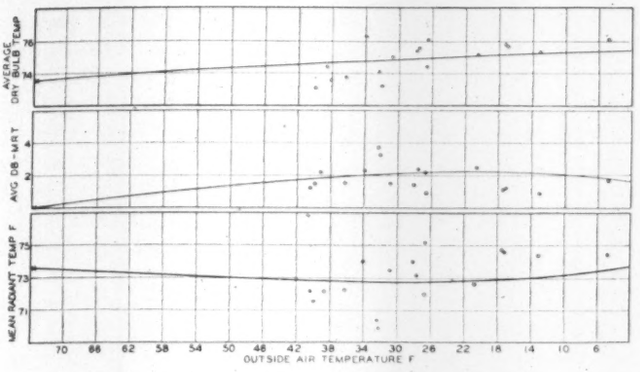

Fig. 8—Relation between outside dry-bulb temperature and mean radiant temperature, average dry-bulb mean radiant temperature difference and average dry-bulb temperature (30-in. and 78-in. levels) for radiation room

(×) dry-bulb at 40 per cent RH and 68 deg ET where subjects were comfortable with no difference between air and mean radiant temperatures

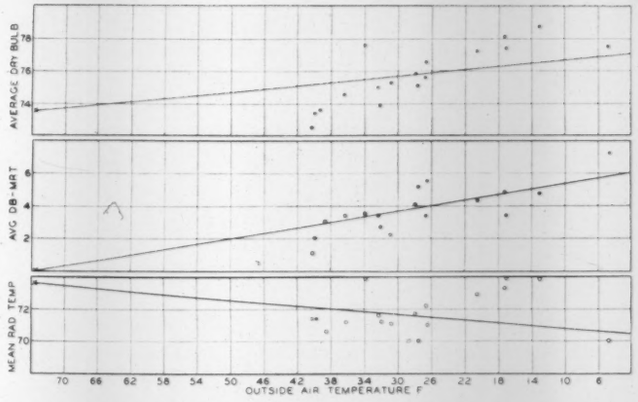

Fig. 9—Relation between outside dry-bulb temperature and mean radiant temperature, average dry-bulb mean radiant temperature difference and average dry-bulb temperature (30-in. and 78-in. levels) for convection room

(×) dry-bulb at 40 per cent RH and 68 deg ET where subjects were comfortable with no difference between air and mean radiant temperatures

of the inside wall surface temperature with decreasing outside temperature. In the convection heated room where there was no compensating radiant effect with decrease in outside temperature, the cooling effect with decrease in outside temperature; that is, the cooling effect of radiation from occupants to the colder walls, required a compensation in the form of higher dry-bulb and effective temperatures.

While the variation of the required indoor effective temperature with outdoor dry-bulb temperature in the two rooms is significant, the exact magnitude, as indicated in Figs. 6 and 7, will naturally vary with the insulating value of the walls of the enclosure, and the percentage of the walls exposed to the outside.

The variations in outside dry-bulb temperature with the mean radiant temperature, the difference between the average dry-bulb temperature within the room and the mean radiant temperature, and the average indoor dry-bulb temperature are all plotted for the radiation and convection rooms in Figs. 8 and 9, respectively. Again, while there is considerable variation in the points so that exact curves cannot very satisfactorily be drawn, definite tendencies are indicated which differ widely for the two rooms. In the convection room there is a definite increase in the difference between the average air temperature and the mean radiant temperature, with lower outside air temperature, while the same difference for the radiation room is either constant, or possibly there is some slight tendency for a decrease in this temperature difference with lower outside temperature. This indicates again that the increased radiation from the warmer radiator in colder weather compensates for the increased radiation loss from the subject to the colder walls and windows, thus accounting for the demand for increasing effective temperature to maintain comfort in the convection room with little or no increase in this demand for the radiation room.

The mean radiant temperature observed in the two rooms is plotted against the effective temperature for ideal comfort in Fig. 10. No relation is shown. Two distinct factors serve to confuse the relationship which might be expected to exist; that is, provided all other environmental factors remain constant, the higher the mean radiant temperature the lower the required effective temperature. These two disturbing factors are: first, the fact that the dry-bulb temperature of the air for a given effective temperature, and therefore indirectly to a lesser extent the mean radiant temperature, increases with decreased moisture content of the air; and second, the fact that there was a small though persistent average greater air movement in the convection heated room than was observed for the radiation room.

The relation between effective temperature for ideal comfort and the difference between the average of the dry-bulb temperatures at the 30- and 78-in. levels, and the mean radiant temperature is plotted in Fig. 11 for both rooms. A rather definite curve is indicated, following the logically expected trend, namely, that with increasing difference between the dry-bulb temperature of the air and the mean radiant temperature, a higher effective temperature is required for comfort. The higher mean radiant temperature and therefore the lower the dry-bulb to mean radiant temperature difference in the radiation room is shown by the segregation between the points for the radiation room and those for the convection room.

Sensations of Draft, Coolness, and Warmth

In stating their sense reactions to the conditions in the rooms the subjects were instructed to consider their feelings as regarding *first,* drafts, *second,* local sensations of coolness (if they did not interpret them as drafts), *third,* local sensations of warmth and *fourth,* any other unusual reactions they might have to the conditions in the room. Two of the subjects either took this too seriously or were unusually sensitive for almost without exception they reported one or more of these unusual sensations. Excluding these two subjects there were seven sensations of draft for 149 *subject-periods* in the radiation room at times when the average subjects indicated feelings of *ideally comfortable* and 11 sensations of draft for 148 *subject-periods* in the convection room. With the same *subject-periods* there were no local sensations of coldness and only one sensation of excessive warmth in the

radiation room, and eight sensations of local coldness, and three sensations of excessive warmth in the convection room.

The two unusual subjects reported 19 local sensations of draft, one of coolness, and none of warmth for 50 *subject-periods* in the radiation room, and 71 local sensations of draft, 36 of coolness, and 11 of warmth in the convection room for 49 *subject-periods*. Inasmuch as the subjects changed the positions in which they sat from test to test, it is not likely that the unusual difference in number of local sensation votes can be attributed to differences in position.

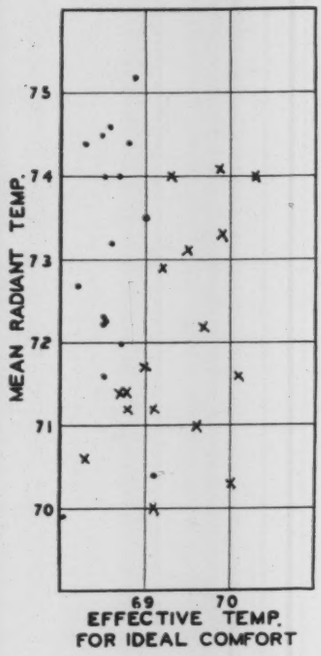

Fig. 10—Relation between effective temperature at which comfort was realized and mean radiant temperature

(·) indicates subjects in radiation room. (×) indicates subjects in convection room

Variation in the Effective Temperature Requirements

The rather high effective temperature, demanded by the subjects for comfort, even on relatively warm days, was somewhat surprising. For example, it will be seen from the curves Figs. 6 and 7, that for outside temperatures in the neighborhood of 40 F, approximately 68.6 deg ET was required in the radiation room, and 69.0 deg ET in the convection room. These requirements did not seem to correlate well with the generally accepted 66 deg ET for comfort in warm wall rooms used currently in THE ASHVE GUIDE 1940. In order to give additional information on this point and in order to give directly comparable requirements for comfort in warm rooms, three extra tests with the same subjects were conducted in the Laboratory's psychrometric chambers following the completion of the tests reported for the radiation and convection rooms.

These three extra tests were similar to those conducted and reported[1] by the Laboratory at earlier dates; but instead of finding the average subject comfortable at from 64-66 deg ET, as observed in the earlier studies, these eight subjects were found to be comfortable in the psychrometric rooms at an average of 67.5 deg ET. The only obvious reasons which may be advanced for this change are either individual differences in the eight subjects from those used in the earlier studies, or a change in the desired indoor temperature for comfort since the earlier studies were conducted.

In the three extra tests made in the psychrometric rooms great care was taken to bring the wall surfaces into equilibrium with the atmospheric condition as had been done in earlier studies. In these tests there were available means for observing the mean radiant temperature which were not had in the earlier investigations. The actual mean radiant temperature found in the psychrometric rooms was about 1½ F above the air dry-bulb temperature at the 30-in. level. This apparently resulted from radiation from the bodies of subjects and observers and from the lighting. While no tests were made, it is probable that under such equilibrium conditions with no subjects or lights in the room there would have been no difference between the air temperature and the mean radiant temperature of the surface.

Radiation and Its Measurement

Since 1920 the ASHVE Laboratory and others in this country have been studying the effect of the temperature, moisture content, and movement of the air upon a person's feeling of warmth. During the same period contemporaries in England led by Leonard Hill, Vernon, and Bedford, have paid a great deal of attention to the effect of the fourth environmental factor, radiation. Perhaps the principal reason for this difference in interest has been the greater attention paid in England to the environmental effects on industrial workers exposed to high temperatures where radiation is of great importance.

The studies abroad have naturally resulted in consideration of methods of measuring and evaluating radiation and the development of the Vernon globe thermometer, which is an instrument designed to give an integrated measure of the radiation effect at a point from the entire 4π radians of solid angle surrounding it. Other instru-

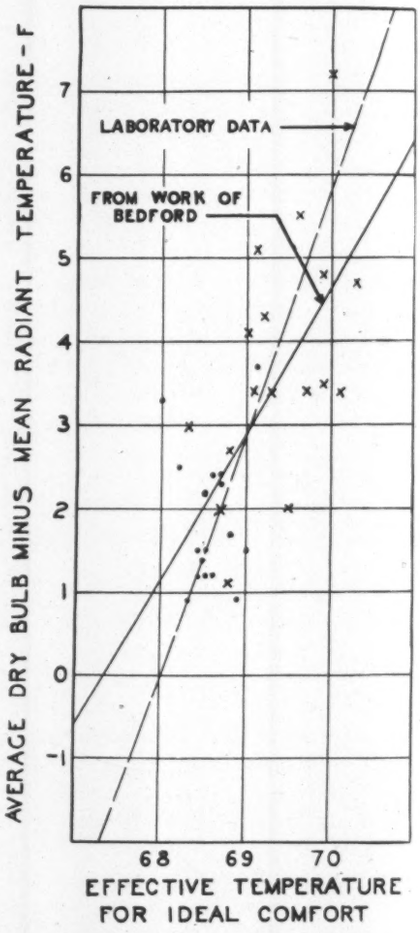

Fig. 11—Relation between effective temperature at which comfort was realized and difference between dry-bulb and mean radiant temperature

(·) indicates subjects in radiation room. (×) indicates subjects in convection room. Solid curve shows this relationship based on work of Bedford and the early work of the Laboratory

[1]See Bibliography 4, 5, 6, and 8.

ments used in the past few years in this country and abroad are the eupatheoscope and the thermo-integrator.[2] The Vernon globe thermometer consists of a thin-shelled copper sphere 6 in. in diameter, blackened on the outside, with the sensitive bulb of a mercury thermometer inserted into the center. This globe is sensitive to radiation, air temperature, and air movement. If it is placed within an enclosure where the inside wall surfaces are above the inside air temperature, the radiation received by the sphere will raise its temperature to a point above that of the surrounding air. This difference between the air temperature and the globe temperature is a function of the radiation received from all surfaces in view of the globe. However, in order to be used to evaluate the magnitude of the radiation effect, a correction must be made for the air velocity near the globe in order to correct for the reduction in the resulting temperature difference.

Thus the Vernon globe thermometer requires not only the temperature of the globe itself but the temperature of the surrounding air unaffected by radiation, determined with a thermometer shielded with bright metal from radiation, and the air velocity near the globe determined by the Kata thermometer.

The use of this instrument at best demands unusual skill. *First,* its blackened surface must approach very nearly that of a true black body. *Second,* the determination of true air temperature unaffected by radiation is difficult and the ordinary method of shielding against radiation by bright metal cylinders is questionable. *Third,* the determination of small air velocities in occupied spaces by the Kata thermometer is a difficult and laborious task.

One inherent difficulty in the use of the Vernon globe thermometer results from the fact that the observer must come in close proximity to the instrument in order to determine the globe temperature, the shielded air temperature near it, and the Kata thermometer observations, thus unavoidably disturbing the very environmental conditions being measured.

The Laboratory has been able to improve the globe thermometer in its adaptability to the problem at hand by applying thermocouples not only for observing the globe temperature but also for observing the air temperature unaffected by radiation. It has been a well accepted fact that a very fine, bright metal thermocouple with no bead at the junction probably gives as nearly the true air temperature unaffected by radiation as is possible by any available methods. Based upon this fact the Laboratory developed an instrument in which a differential thermocouple of No. 36 copper-constantan wire was used, with its eight junctions alternately placed within the globe and in the air at its four sides; thus giving directly and more sensitively the temperature difference between the globe and its surrounding air. The use of one of these junctions as a single couple served to give the true globe temperature and that of another the true air temperature. This arrangement had the added advantage that observations could be made by means of the potentiometer located in the observation room. The use of a sensitive hot-wire anemometer developed by the John B. Pierce Laboratory made it possible to read the surrounding air velocity remotely; that is, its sensitive element was placed near the globe and read in the observation room.

The importance of the factors in the use of this instrument, previously described, is indicated by the fact that 100 per cent differences in resultant observations were had at times between the standard Vernon globe and the Laboratory instrument. The greater variability in the results in the older instrument together with the fact that the improved instrument always gave the greater resultant effect (any errors due to imperfection in black body surface, and due to the effect of radiation on the true air temperature would tend to make the reading too small) indicates that these improvements resulted in more accurate evaluations. Such possible variations indicate clearly that further study should be given to desirable instruments and methods for their use in observing mean radiant temperatures.

Fig. 12 shows the standard Vernon globe thermometer with the shielded thermometer, Kata thermometer, and bare air thermometer at the reader's right, and the instrument developed by the Laboratory with the Pierce hot-wire anemometer at the left.

British usage has developed the term *mean radiant temperature,* which is assumed to be the integrated average temperature of all surrounding surfaces from which radiation may come. Otherwise defined, mean radiant temperature is meant to be that uniform temperature to which all surfaces in view of a point must be heated in order to give the same resultant effect.

While the mean radiant temperature thus defined and measured by the globe thermometer is the best expression of the effect of radiation at a point available at the present time, there are criticisms of its use in defining the environmental effect of radiation for the reason that a person is sensitive to the distribution of radiation from various directions, as well as to the total effect. As an example, a person would undoubtedly feel very different in the following two conditions, both of which might give the same mean radiant temperature; *first,* a condition where the given mean radiant temperature is the result of all the

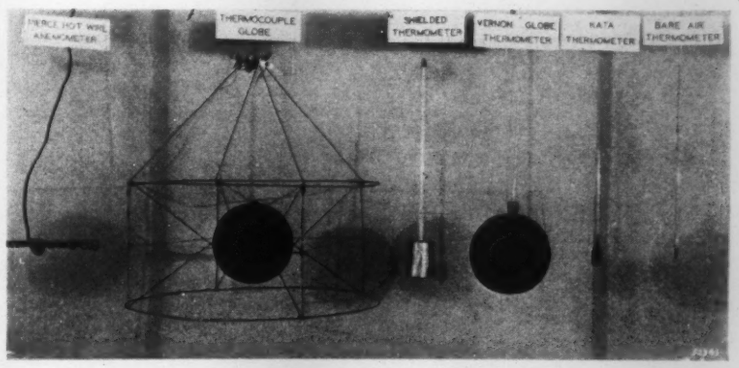

Fig. 12—View of instruments for measuring air environment

[2]See Bibliography 12.

surfaces in view of the point under consideration being at one temperature; and *second*, a condition where the same mean radiant temperature resulted from a very high temperature source in one direction (such as a large steam-heated radiator), and a low temperature source in the other direction (such as a large single-glazed window on a sub-zero day). It is probable that a directional instrument consisting of a single radiation measuring cone which might be oriented at will, or a single instrument having six equal segments, taking in practically the entire surrounding solid angle, would serve admirably. While the combined effect given by the six segments of this instrument would neither give directly the mean radiant temperature, nor as complete an integration of the combined enclosure as the globe, it would probably be sufficiently accurate for the purpose, and would at the same time give the individual directional effects which are probably of greater value in our air conditioning practice than are the over-all effects, particularly in this country where occupants of air conditioned spaces seem to be sensitive to directional feelings of warmth or coldness.

A little attempt was made in this study to measure these directional effects by determining the contribution of each surface in the room to the mean radiant temperature as measured by the globe thermometer. A test was conducted in the rooms without occupants when the mean radiant temperature was measured by the globe thermometer, and the temperatures of all surfaces in the rooms were measured by both thermocouples and a cone-shaped radiation meter. The results of this test are given in Table 4. For the radiation room Column *A* gives the surface under consideration: *B*, the solid angle, expressed in radians, subtended by the surface from a point at the center of the Vernon globe thermometer; *C*, the temperature of the surface in question as observed by thermocouples; and *D*, this same temperature expressed in degrees absolute and raised to the fourth power. Column *E* gives the product of the solid angle and the surface temperature; and *F*, the product of the solid angle times the surface temperature expressed in degrees absolute and raised to the 4th power. Columns *C'*, *D'*, *E'*, and *F'* give similar values observed in the convection room.

The solid angles, Column *B*, were evaluated in radians and their sum should equal the entire 4π radians in the solid spherical angle. Actually, the integration gave 12.563 or 3.99π radians. While the mean radiant temperature in the radiation room as observed by the globe thermometer was 69.3 F, the integrated average surface temperature resulting from summing up Column *E* was 65.4 F. The calculated mean radiant temperature based upon the fourth power of the temperatures was 67.9 F. For the convection room a similar integration is indicated. It is worthy to note that in the convection room where the surfaces were at more nearly the same temperature there is no appreciable difference between the calculated mean radiant temperature and the calculated integrated average temperature; while in the radiation room the presence of radiators approximately 60 F hotter than the integrated average temperature raised the calculated mean radiant temperature 2.5 F above the calculated average temperature.

Effective Temperature — Mean Radiant Temperature Relationship

It should naturally be assumed that a certain definite relationship should exist between the comfort of occupants in an enclosure and the mean radiant temperature, provided all other environmental factors remain unchanged. Closer correlation between the work of the Laboratory and our English contemporaries over the past twenty years should have made this relationship obvious. A careful study of the work of Bedford and his associates in England in relation to the work of this Laboratory makes such a comparison possible, although arrived at indirectly.

Bedford and others have indicated in their published results a relation between mean radiant temperature and dry-bulb temperature for the same feeling of warmth. From his work, therefore, it is possible to determine the necessary elevation or depression in the dry-bulb temperature to compensate for a counter depression or elevation in the mean radiant temperature observed in an enclosure, while from

Table 4—Relation Between Mean Radiant Temperature Determined by Globe Thermometer and Calculated from Surface Temperatures and Solid Angles Subtended by These Surfaces

A	B	Radiation Room				Convection Room			
		C	D	E	F	C'	D'	E'	F'
Surface	Solid Angle Radians	Surf. Temp.		B x C	B x D	Surf. Temp.		B x C'	B x D'
		°F	T⁴			°F	T⁴		
S. E. Wall	2.395	64.3	756x10⁸	154.0	1810x10⁸	64.4	757x10⁸	154.2	1815x10⁸
S. W. Wall	0.870	64.0	754	55.6	655	65.4	761	56.8	662
Ceiling	3.274	64.8	761	212.2	2491	63.2	747	207.0	2446
Floor	2.759	67.8	775	187.0	2138	67.0	771	184.8	2126
N. E. Window	0.194	53.6	695	10.4	135	46.2	655	9.0	127
N. E. Rad.	0.062	124.0	2189	7.7	136	67.0	771	4.2	48
N. E. Wall	0.614	62.9	747	38.6	459	64.1	754	39.0	463
N. W. Window	0.302	51.3	683	15.5	206	45.8	654	13.8	197
N. W. Rad.	0.078	125.5	2207	9.8	172	66.8	770	5.2	60
N. W. Wall	2.015	64.5	756	130.3	1623	63.9	752	129.0	1520
Totals	12.563[a]			821.1	9725x10⁸			803.0	9464x10⁸

$\frac{821.1}{4\pi} = 65.36$ F Avg surface temp

$\sqrt[4]{\frac{9725 \times 10^8}{4\pi}} - 460 = 67.9$ F—MRT

Observed MRT = 69.3 F

$\frac{803}{4\pi} = 63.92$ F Avg surface temp

$\sqrt[4]{\frac{9464 \times 10^8}{4\pi}} - 460 = 64.0$ F—MRT

Observed MRT = 67.5 F

[a] $12.563 = 3.99\pi$ Radians

Table 5—Relations Between Positive or Negative Changes in Mean Radiant Temperature and Necessary Change in Dry-Bulb or Effective Temperature to Maintain the Same Feeling of Warmth[a]

CONDITIONS														
	E.T.		66	66	66	73	73	73	80	80	80	85	85	85
	R.H.		20	50	80	20	50	80	20	50	80	20	50	80
	D.B.		73	70	67.5	83.5	79	75	95.3	88	83	105	95	88.5
CHANGES	ΔMRT	ΔDB	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET
	-1	.98	.63	.66	.68	.53	.56	.60	.43	.47	.51	.37	.41	.46
	-3	2.7	1.75	1.81	1.92	1.46	1.55	1.65	1.19	1.30	1.43	1.05	1.18	1.27
	-6	5.2	3.36	3.48	3.67	2.80	3.00	3.19	2.29	2.50	2.72	1.97	2.19	2.44

[a]Relation between mean radiant temperature and dry-bulb temperature from Bedford's work. Effective temperature relationship based on laboratory's work.

the work at the Laboratory it is readily possible to determine the elevation in the effective temperature corresponding to a given rise in the dry-bulb temperature. Combining these two relationships, Table 5 is presented, giving the elevations or depressions in effective temperatures to compensate for given depressions or elevations in mean radiant temperatures of 1, 3, and 6 F. These comparisons are given for a range of from 66 to 85 deg ET, and for three relative humidities, thus including the most important variations in atmospheric conditions found in heating and cooling practice. While these comparisons are given for the necessary *increase* in the effective temperature to compensate for the depression in mean radiant temperature, the same comparison to within the limits of accuracy of the original data applies when the effective temperature of the surroundings is *decreased* to compensate for an *increase* in the mean radiant temperature.

By interpolation it is found that for a condition of 70 deg ET and 30 per cent relative humidity an elevation of 1 deg ET will about compensate for a 2 deg lowering of the mean radiant temperature. For the same effective temperature and 20 per cent relative humidity the same change in mean radiant temperature requires a little smaller change in the effective temperature, while with 80 per cent relative humidity a greater change in the effective temperature is required for compensation. Although there is some variation in the compensations in effective temperature required for a given change in mean radiant temperature, it will be observed from the table that these variations are not great. In the column ΔDB the variation in the dry-bulb temperature as taken from Bedford's work is shown. For example, a depression of mean radiant temperature of 1 degree is compensated for by an elevation in the dry-bulb temperature of 0.98F. In this study it is evident that both the relationship and the measurement of the mean radiant temperature should be further studied as factors in a person's sensation of comfort.

These relationships are somewhat more readily shown in Fig. 13, where the relation between the positive or negative change in mean radiant temperature, and the necessary negative or positive change in effective temperature to result in no change in the feeling of warmth is plotted over a wide range of effective temperatures and for three different relative humidities. These curves are based upon Table 5.

Applying the information from Table 5 and Fig. 13 to the data plotted in Fig. 11, the solid line represents the relationship between the effective temperature for comfort and the dry-bulb, mean radiant temperature difference which the correlated work of Bedford and this Laboratory would predict. The slope of this curve is obtained from interpolation on Fig. 13 and the curve itself is so placed in Fig. 11 that it passes through the point 69.0 deg ET and 2.9 F which is the algebraic mean of all the plotted points. Its agreement with the data is fair although the curve as drawn from the points fits the data better.

From Fig. 5 it will be observed that 2.3 deg ET elevation was required in the test in the radiator heated room on January 19 to result in an average change in the subjects' index of comfort from *4* or *ideally comfortable*, to either *3* or *5*, *comfortably cool* or *comfortably warm*, respectively. Columns L_r and L_c, Table 2, indicate that for the different tests the necessary change in effective temperature to result in a change in comfort index from *4* to *3* or *5* ranges from 1.9 to 3.7 deg ET for the radiation room, and from 1.7 to 4.1 deg ET for the convection room; or an average of 2.7 deg ET change for either room. The first Laboratory study[3] made in 1923 indicated an optimum comfort at 65 deg ET and a required change of 1 deg ET for a comfort change from *3* to *4*. Later Laboratory studies[4] show necessary changes varying from 2.3 to 4.3 deg ET for a comfort change from *3* to *4*, agreeing well with the results of this study. This relationship between the necessary change in effective temperature

[3]See Bibliography 4.
[4]See Bibliography 5, 6, and 8.

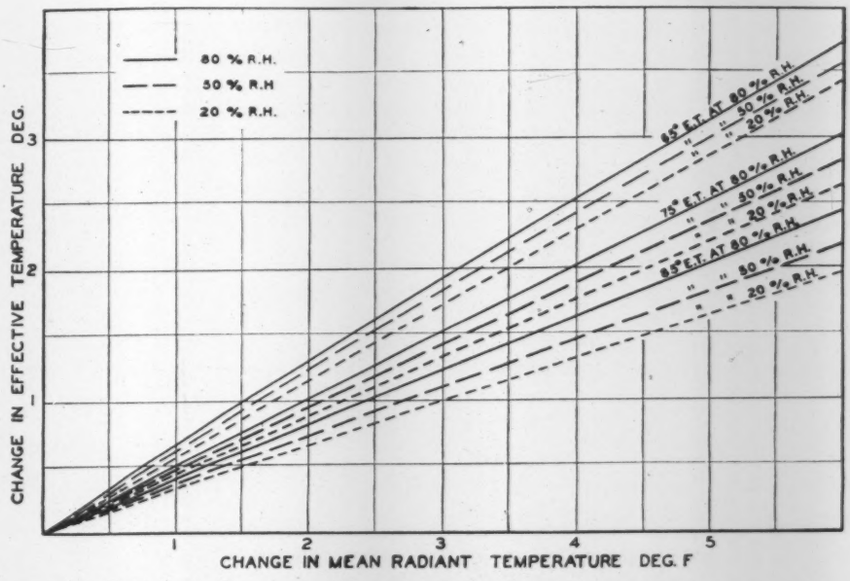

Fig. 13—Relation between positive or negative change in mean radiant temperature and the necessary negative or positive change in effective temperature resulting in no change in feeling of warmth

for a change in unit index of comfort is closely tied up with the impression which is conveyed to the subjects regarding what they are to interpret as, *comfortably cool,* or *comfortably warm.* Throughout the work of the Laboratory an attempt has been made to have the subjects interpret *comfortably warm* (or *comfortably cool*) as a condition in which they are essentially comfortable but one in which, if a change were required, would be made in the direction of *ideally comfortable.* It is probable that a finer interpretation of this difference was made by the subjects in the earlier Laboratory studies than has been made since.

Considering a change of 3 deg ET as necessary to account for a change of one unit in the comfort scale, it will be observed from Table 5 and Fig. 13 that if no other factors in the environment than the mean radiant temperature were changed the required elevation in mean radiant temperature to result in a change from ideally comfortable *4* to *5* would be about 6 deg with some variation depending on the temperature and the humidity of the condition. This fact, together with the relatively small difference between the average dry-bulb temperature and the mean radiant temperature in either room, and more particularly in the radiation room, shows clearly that the effect which it was attempted to evaluate in this study was little more than above the threshold of observation and accounts amply for the fact that while this study indicates definite trends when the average results are plotted the small effects could not be very accurately evaluated. Since in this study the mean radiant temperature effects which it was attempted to evaluate were exaggerated over and above what they would be in normal enclosures, *first,* by building these rooms of relatively poorly insulated walls (having a transmission coefficient, U, of 0.19) and *second,* since these rooms had four almost full directions of exposure, rather than the usual one or two exposures found in most enclosures, it will be seen that the small effects observed were large when compared with what should be expected in normal rooms.

In this connection it is of further interest to note that the effect of the radiation was always such as to reduce the difference between the air temperature and the mean radiant temperature. In no case in the rooms in which this study was made did the mean radiant temperature exceed the air temperature, even in tests 18 and 19 when the windows were entirely blanked off in the radiation room with 1 in. structural board insulation so as to reduce to a minimum radiation to cold windows.

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FEDERAL GOVERNMENT RECRUITS DRAFTSMEN

Engineering draftsmen in various optional fields, including heating and ventilating, are urgently needed by the United States Government. A civil service examination held last fall failed to produce enough eligibles to meet the demand of the national defense program. The United States Civil Service Commission, Washington, D. C., has, therefore, reannounced the examination and will accept applications until December 31, 1941. The salaries of the positions range from $1,620 to $2,600 a year, less a 3½ per cent retirement deduction.

Competitors will not be required to take a written test, but will be rated on their education and experience as shown in their applications, and on corroborative evidence.

Further information and application forms may be obtained from the Secretary of the Board of U. S. Civil Service Examiners at any first- or second-class post office, or from the U. S. Civil Service Commission, Washington, D. C.

BOOK ON TEMPERATURE CONTROL ANNOUNCED

A book on *Temperature—Its Measurement and Control in Science and Industry* is now available. It contains valuable papers presented at a Symposium sponsored by *American Institute of Physics* in cooperation with *National Bureau of Standards* and *National Research Council,* held last winter in New York. It is especially valuable to medical men, electrical, heating, mechanical and refrigerating engineers, and plant operating officials.

Comfort in High Altitude Flying

By D. W. Tomlinson,* Kansas City, Mo.

SUMMARY—The developments of the past decade in heating and ventilating airplanes are reviewed. The first cabin steam heating system is described and details are given of the five related systems that are now used for providing comfort in the modern Stratoliner for high altitude flying. The details of operation at altitudes between 8,000 and 19,000 ft are given, and the methods of control, the cabin supercharging system, and the air distribution are also described

DURING the past 10 years the improvement that has been made in the airplane as a whole is almost phenomenal. The heating and ventilating system of passenger planes has always presented a major engineering problem, and the problem has not yet been completely solved. However, all air travelers will agree that the safety and comfort provided for the present day airline passenger has been greatly improved, and the contrast between the heating and ventilation of the modern Stratoliner of today and the aircraft of 10 years ago shows that some notable advances have been made.

Ten years ago heating and ventilating was considered of more or less secondary importance. The supply of cold air was taken in through small *horns* located at each seat, these horns extending through the fuselage to the outside. They were designed so they could be turned to take in the desired amount of air or turned up or back to stop the flow of air. Naturally, considerable noise, vibration, and quite often fumes from the engines were brought into the cabin, but at that time it was considered by most people as just part of air travel. No spent air or ventilation valve was provided to help control the amount of foul or used air escaping from the cabins. Leaks around doors and windows, which were usually plentiful, were depended upon to solve this problem.

The conventional heating system in airplanes of 10 years ago usually consisted of a metal shroud covering the exhaust tail pipe as shown in Fig. 1. This tail pipe was ordinarily on the under side of the plane and quite often extended almost the full length of the cabin in order to obtain sufficient heat. The air entered the duct thus formed at the forward end between the exhaust pipe and the shroud and after passing along the hot exhaust pipe, was admitted to the cabin through adjustable sliding valves in the aisle between the seats, or through tubular ducts along each wall near the floor as indicated in Fig. 1.

Admittedly poor circulation, and quite often inadequate heat, were obtained with this arrangement, yet it was many years before it was changed. The admittance to the cabin of exhaust gases and carbon monoxide from breaking exhaust pipes and loose connections was always feared, and now it is sometimes wondered how the airlines operated as long as they did without more serious trouble from the hazard of asphyxiation and fire in the cabin heating systems of that period.

In 1933, when the Douglas DC-1 was built for TWA, the first cabin steam heating system, to the author's knowledge, was installed for airline use. The system was of the low-pressure type, operating under only a few pounds pressure. The water entered the boiler at the bottom by gravity feed and went out the top as steam to the steam radiator, which was located in the fuselage below the cabin floor. The boiler was located in the exhaust manifold in one engine nacelle which is the streamlined compartment housing the engine on the wing. Cold air picked up at the nose of the plane was directed by an electrically operated automatic temperature control either through the steam radiator or around it, depending upon the cabin heat demand. Fig. 2 shows the heating and ventilating system of the DC-2, which is identical with that of the DC-1. After this first DC-1 steam heating system, fast strides were made to the present-day heating and ventilating system now installed in the Stratoliner, designed and built by Boeing.

The heating and ventilating system of the Stratoliner consists actually of five separate, though intimately related, systems, as follows: (1) Fresh Air System, (2) Spent Air System, (3) Warm Air System, (4) Steam System, and (5) Ground Air Conditioning System.

Fresh air is supplied to the cabin by means of an air scoop on the upper side of the fuselage through which air is forced by the forward speed of the airplane into a distribution duct which extends the full length of the passenger cabin and which is located on the ceiling as indicated in Fig. 3. This is divided longitudinally into two sections, the lower of which contains grilles through which air is distributed directly into the cabin while the upper section is a supply duct for the individual outlets which are located at each seat and upper berth. The individual outlets are also provided in the ladies' and men's dressing rooms and other sections of the plane forward of the passenger cabin.

After the air enters the scoop, it passes through a centrifugal water

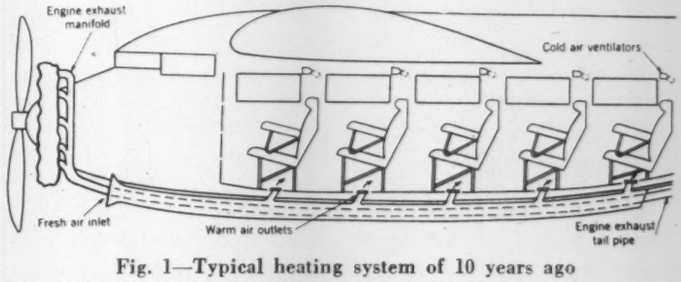

Fig. 1—Typical heating system of 10 years ago

*Vice-President in charge of Engineering, Transcontinental & Western Air, Inc.

Presented at the 47th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., January, 1941.

separator device which is an integral part of the duct just inside the fuselage skin, and which is intended to prevent water from being forced into the distribution duct in the cabin. There are two valves located in the outlet duct behind the water separator, the foremost one being used for closing the air inlet during the time the cabin is supercharged or during extremely cold weather, when the outside cold air is not necessary. It is operated from the cockpit by means of a lever on the floor behind the seat occupied by the First Officer. The rearmost valve controls the flow of air through the lower section of the ceiling duct only and can be operated only from the control on the front bulkhead in the passenger's cabin. Thus the Flight Engineer, who controls the cabin supercharging system, determines the source of all air entering the cabin but cannot control the distribution of it; and conversely, the hostess in the cabin determines the distribution of the air but cannot control the source.

The individual ventilator nozzles located in the wall next to each seat and within each berth, have both a directional and a volume flow adjustment which is formed by a ball-and-socket made of sound-deadening plastic material. Each nozzle may be adjusted to release a high velocity air jet in order to obtain maximum air recirculation effect. A rubberized fabric hose connects the individual ventilators to the main supply duct and is concealed behind the cabin lining.

The spent air is exhausted into the lower fuselage section below the cabin floor through vents in the floor, which are located under the seats and so arranged as to give even distribution throughout the cabin. The air is discharged from the airplane through either an auxiliary outlet valve, or the supercharger control units or both. The auxiliary outlet valve is located in the lower covering of the airplane just aft of the front spar bulkhead in the accessory compartment and is manually operated by the same lever which is used to operate the fresh air inlet valve.

Cabin Supercharging and the Warm Air Supply

The pressure, air supply, control, and temperature conditioning system is provided in duplicate, there being one system on each side of the airplane, (Fig. 4) either system being capable of maintaining satisfactory cabin temperature and pressure should the other system fail.

All doors and hatches open inward and, therefore, are sealed against rubber gaskets under the influence of cabin pressure. The skin or fuselage covering continuity is not broken at the intersection with the center section of the wing and forms a cell which is circular in section, cigar-shaped in form. There are 17 passenger compartment windows measuring 12 in. high x 16 in. wide, the panes of which are a single-ply section of $\frac{5}{16}$ in. Plexiglas. Plexigas was chosen for its high strength-weight ratio. The

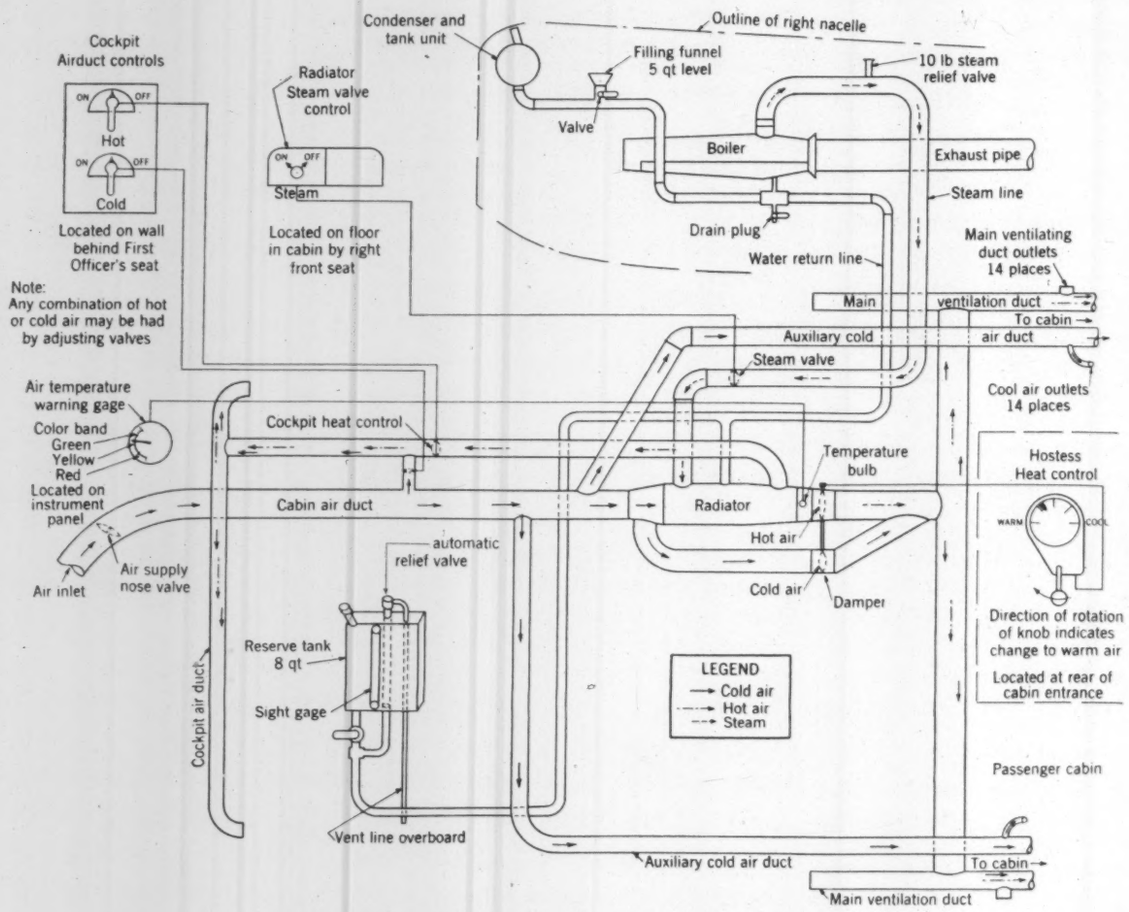

Fig. 2—Heating and ventilating system of DC-2 plane

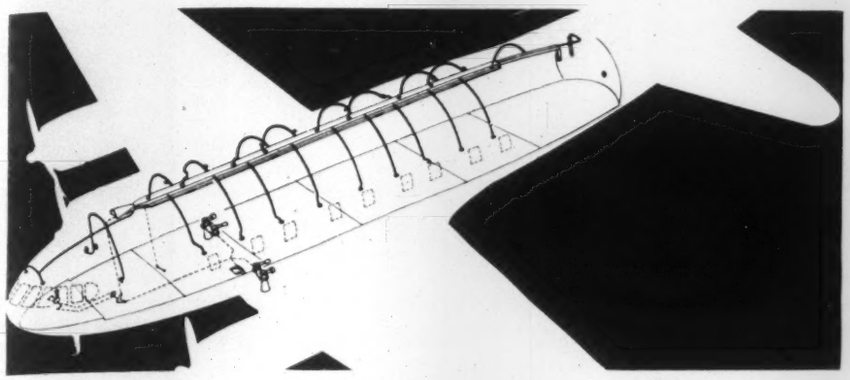

Fig. 3—Fresh air system

cabin is designed to withstand 6 lb per square inch internal pressure, although the maximum internal working pressure intended for transport operation at present optimum altitudes is but 2.5 lb per square inch.

The cabin floor level is located below the center line of the fuselage at its largest section. The cargo compartments and the accessory compartment between them under the floor are maintained under full cabin pressure and are part of the circulatory system. The fuselage is pressurized only to the rear end of the cabin and at that point, there is a removable circular hatch in one side of the hemispherical bulkhead to enable inspection of the interior of the tail of the airplane. An emergency air valve set to relieve at a pressure of 2.65 lb per square inch is also located in the bulkhead as a safeguard against building up excessive internal pressure. This valve has a capacity equaling the total overloaded discharge of both cabin superchargers at sea level and during maximum supercharger speed, without causing the cabin pressure to exceed 3.6 lb per square inch.

This airplane carries a Flight Engineer as part of the flight crew, one of his duties being to handle the cabin supercharging. His station is behind the co-pilot and he has an instrument board in the side wall of the flight compartment on the right hand side of the cockpit as shown in Fig. 5. The pressure-cabin instruments are mounted on this panel in a group by themselves, separated from the surrounding instruments. In this group and at the lower left is a supercharger discharge pressure gage and a two-way selector valve for connecting the gage to the right or to the left supercharger in order that the pressure rise may be known. On the right is a suction gage with a three-way selector valve connected to it. When the latter valve is in the central position, the suction gage indicates cabin pressure referred to outside pressure. When the valve is in the left or right position, the suction gage indicates depression in the throat of the flow-measuring venturi of the right or of the left pressure air supply system. A selector valve is located in the uppermost portion of the panel for the purpose of manually selecting operation on either the right or left outlet valves, or both as desired. Below this space, a vertical speed indicator, referred to cabin pressure instead of to atmospheric pressure, indicates the rate of cabin pressure change in terms of feet per minute. Next to this gage is an apparent altitude green warning light to inform the engineer of the absolute pressure in the cabin. The green light can be pre-set by a knob below it to shine at consecutive apparent pressure altitudes of 8000, 15,000 and 18,000 ft.

The air to be compressed is drawn from an inlet in the leading edge of the wing adjacent to the wing fillet. From there it passes through a water separator, through the supercharger (Fig. 6) which is driven by the engine, through the steam radiator in the leading edge of the wing, through the intercooler to reduce the temperature of the air, through the inflow control regulator, on through the cabin, out of the cabin through the floor into the compartments under the floor and then through the outflow valve which is incorporated in the unit containing the inflow valve.

If the cabin temperature is too cold, feedwater pumps supply water to the boilers and these in turn supply steam to the radiators, thus raising the temperature of the incoming air passing through the radiator. If the cabin temperature is too high, the feedwater pumps do not operate, consequently the air passes through the cold radiator and the intercooler. On warm days during supercharged operation, the supercharger discharge temperature is such that it is necessary to operate with the intercooler valves open for maximum cooling.

The inflow of pressure air is fixed by the automatic controls at an impact value corresponding to the impact produced by 250 cfm per blower at sea level. Except for changes due to cabin leakage, the flow value is unaffected by changes of other conditions, such as blower speed. The absolute pressure in the cabin is maintained substantially constant between 8000 ft and 15,000 ft pressure altitude. At altitudes higher than 15,000 ft, the differential pressure of the cabin referred to the atmosphere is automatically held at approximately 2½ lb per square inch, resulting in an apparent cabin altitude of 11,000 ft at an actual alti-

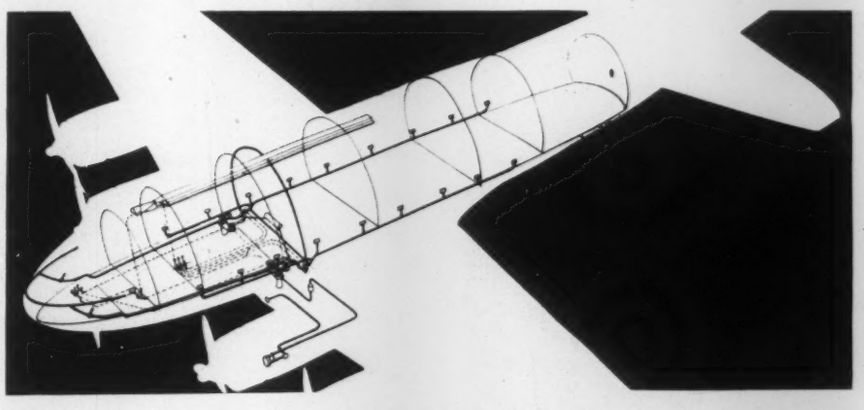

Fig. 4—Hot air and supercharging systems

tude of 19,000 ft (see chart Fig. 7).

The pressure air flows into the spherical casting of the cabin supercharger control and then rises past an inlet flow control valve and flows horizontally into a measuring venturi to be distributed to the cabin. This inlet valve, incidentally, acts

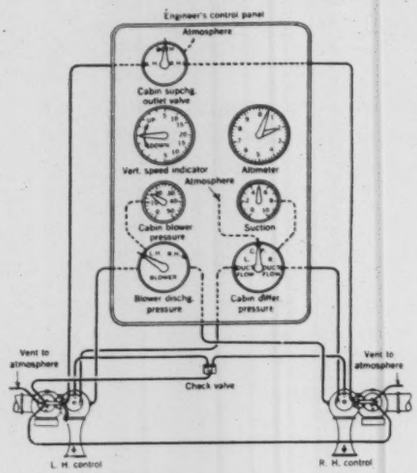

Fig. 5 — Instrument and supercharger control unit connection diagram

as an outflow check valve if the air flow ceases. It may also act as an inflow relief valve if the cabin pressure should tend to fall below the atmospheric pressure on the outside of the cabin.

Air to be exhausted from the pressurized area passes through a screen and then escapes to the atmosphere past an outlet valve which is also a part of the supercharger control unit. The seat of the outlet valve is extended downward through the inlet duct, forming a diffuser; in this manner the seat and diffuser of the outlet valve are heated by the incoming air from the superchargers to prevent formation of ice which might form under the cooling influence of the expanding outflowing air. The outlet valve also acts as an inflow relief if the cabin pressure tends to fall below the atmospheric pressure on the outside of the cabin.

The inlet valve and its motor mechanism are identical in construction to that of the outlet valve, both being of the tulip poppet type, although each motor mechanism is regulated by different control devices. The valve stems are directly attached to reciprocating air motor pistons. The valves are spun from stainless steel sheet metal and the pistons from magnesium, which is also used in the control unit castings. The stainless steel valve stems are guided by means of self-lubricating bronze bushings which space the pistons from the cylinder walls with a small clearance.

Regulation of the inlet valve position is accomplished through adjustment of the intensity of action of the pressure air upon the motor piston. The regulating mechanism is supposed to hold the poppet valve to within about one-thousandth of an inch of the position required by this sensitive mechanism. Since the inlet valve motor is affected by the depression in the flow-measuring venturi, the cabin supercharger serves merely as a ventilating fan when the cabin is not being supercharged. However, when the auxiliary or fresh air ventilation inlet and the auxiliary outlet are closed, the inlet valve motor becomes partially responsive to the position of the outflow valve so that there may be some flow compensation in proportion to the magnitude of cabin leakage as indicated before the inlet valve is designed to serve as an outflow check if the pressure in the supply system falls below that of the cabin.

The air motor of the exhaust or outflow valve is affected by an evacuated bellows which is sensitive to absolute pressure in the cabin and to a piston which is responsive to the differential pressure between the cabin and the atmosphere; hence this is the cabin pressure controlling medium.

During hot weather it is preferable to divert the incoming air from the superchargers into the fresh air ventilation duct in the cabin ceiling. Thus, the entire supply of air is thereby made available at the overhead grilles and individual ventilators, from which it provides the most cooling effect. The second optional flow course consists of a series of ventilation grilles in the side of the cabin near the floor to provide better warm air distribution during cold weather. During average weather the air may be directed to both the overhead and to the floor duct systems or either one. If necessary, the major portion of the heated air can be forced into nozzles extending along the lower edge of the pilot's compartment windshield to defrost the windshield.

There is a cross-over duct between the right hand and left hand systems to insure the even distribution of air throughout the cabin in the event that an inboard engine or supercharger is stopped. Each of the two supercharging systems is capable of providing sufficient heated air to warm the cabin during the most extreme weather conditions and is capable of supplying a quantity of air well in excess of the requirements of the passengers. It is further able to maintain the normal cabin pressure events even at the maximum operating altitudes of the aircraft. The pressure control unit prevents the superchargers from overloading during normal operation regardless of the speed of the inboard engines; however, if one engine stops, the supercharger on the other engine is automatically allowed to overload if additional flow is required to maintain cabin pressure.

The cabin superchargers (Fig. 6) are centrifugal compressors having integral speed-increasing gear boxes. There is one mounted on the rear of the firewall of each inboard nacelle and they are each driven by a flexible shaft from the generator drive of the engine. The superchargers are G. E. Type B-1,

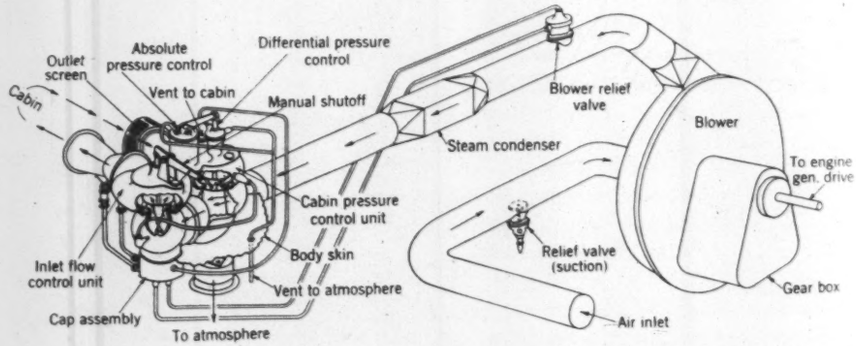

Fig. 6—Cabin supercharging control instrument

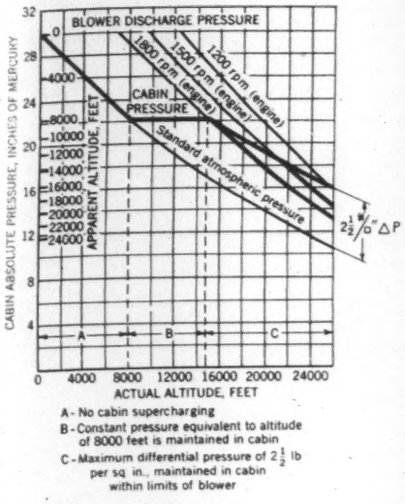
BLOWER DISCHARGE PRESSURE
1800 rpm (engine)
1500 rpm (engine)
1200 rpm (engine)
CABIN PRESSURE
Standard atmospheric pressure
2½ lb/in² ΔP
CABIN ABSOLUTE PRESSURE, INCHES OF MERCURY
APPARENT ALTITUDE, FEET
ACTUAL ALTITUDE, FEET
A
B
C
A - No cabin supercharging
B - Constant pressure equivalent to altitude of 8000 feet is maintained in cabin
C - Maximum differential pressure of 2½ lb per sq. in., maintained in cabin within limits of blower

Fig. 7—Cabin pressure chart

which is rated at 225 cfm at an absolute pressure of 9.35 PSI and a temperature of 70 F at an engine speed of 1500 rpm. The gear ratio between the flexible drive shaft and the impeller shaft is 7.75:1 and as the generator shaft is geared to 11.625 times engine speed, the impeller speed at 1500 rpm engine speed is 17,500 rpm.

A characteristic of all centrifugal blowers is that for any given set of conditions there is a minimum flow required for stable operations. It would, therefore, be impractical to throttle the discharge flow below this critical value without providing some means of relieving the discharge duct pressure in order to prevent surging. A blower relief or surge valve was designed to serve this purpose and is located in the discharge duct just behind the blower. The surge valve is responsive to flow or dynamic pressure at a chosen location in the system, also a certain static pressure in the blower discharge duct is necessary to cause opening of the valve. Without the action of the flow controls, the valve would open as soon as the static pressure in the duct became great enough to lift the piston valve against its weight, as the upper side of the valve would then be exposed to atmospheric pressure. Such action would, of course, be undesirable because high duct pressure is essential to normal operation, regardless of flow. The diaphragm attached to the sliding valve in the cylinder head is subjected on the lower side to cabin pressure and on the upper side to inlet venturi suction. A suitably large venturi depression, that is, sufficient flow for stable operation, will force the diaphragm and valve upward against the downward action of the spring and thus close the vent from the upper side of the piston valve to the atmosphere. Air then leaks through a small orifice at the center of the piston valve and both its upper and lower surfaces are exposed to duct pressure. The valve is thus held closed because the effective area above it is larger than that below it.

An inward opening valve is located in the supercharger inlet duct within the wheel-well of each inboard nacelle. This is designed to open at a suction pressure of ½ lb per square inch within the inlet duct and provide a source of air for the blower in the event that the opening at the leading edge of the wing is closed by ice or snow. The Boeing airplane has four engine nacelles and the two nacelles located nearest the fuselage are called the inboard nacelles.

An automatic temperature control, which consists of an invar rod, is located in a position in the cabin to receive a sample of the air and average cabin temperature as the outlet air passes through the louvers in the cabin floor. This invar rod, through expansion and contraction, operates a micro-electric switch; this in turn puts into action an electric solenoid which operates a valve connected to the plane's hydraulic system. A hydraulic pump in turn operates the feedwater pump that supplies water to the boiler.

The regulation of heat output is accomplished through thermostatic control of the speed of the feedwater pump.

The water supply for each of the duplicate heating systems is about one pint, and is contained in a sump located below the exhaust pipe of each inboard engine. This position is necessarily the lowest point in the system so that all of the water will drain back to the sump when the engines are shut down. Due to the conical shape of the sump, freezing of the contents will not damage it. When the engines are started, heat from the exhaust gases is transmitted to the sump and produces sufficient heat to melt any ice and enough pressure within to force the water out into the circulating system. As the steam rises through the tubes in the system, trapped air is compressed and increases the pump pressure. An airdome, located on the feedwater pump, provides a means of relieving this pressure by acting as an expanding chamber. Excessive pressures are controlled through a pressure switch which automatically cuts out the solenoid circuit to the pump; the circulation in the system then stops until the pressure falls off.

The boiler is a finned cylinder of heat resistant alloy, there being two inserted in the exhaust pipe of each inboard engine to extract heat from the exhaust gases. A threaded spindle is located within the bore of the cylinder. The feedwater pump forces the water along the thread and into contact with the inner surface of the bore at very high velocity, forming steam. The centrifugal force separates any water from the steam being formed, keeping it against the hot surface. The use of fins on the exterior of the boilers provides large heat uptake areas in a small space. As a consequence, there is a very high heat transfer rate although the boiler is very light and compact. Each of the individual heating units has two boilers, each boiler being rated at 40,000 Btu.

The steam generated by the boiler passes through the sump and is admitted to the upper header of the condenser, or radiator, core and distributed from the header to ten banks of corrosion resistant tubes. The heating of the air is naturally accompanied by condensation of the steam, and this condensate is drawn from the bottom header of the condenser by the feedwater pump. The described condenser is highly efficient and yet rugged. It is pressure tested at 1000 lb per square inch, although the maximum steam pressure is limited by the automatic heat controls to 125 lb per square inch.

The ground air conditioning equipment consists of a self-contained unit of the direct expansion type, and is capable of furnishing approximately 55 F. Attachment fittings for the ground air conditioning apparatus are provided on the lower right side of the airplane

just behind the wing fillet trailing edge and are covered by a hinged door. A duct located under the floor, connects with another vertical duct in the rear of the cabin, which serves to connect the ground air conditioning duct with the fresh air distribution duct on the ceiling. Conditioning air is thereby admitted to the cabin through these special ducts to the fresh air system ducts, and from there is exhausted into the cabin either through grilles in the ceiling duct, or the individual ventilators at each seat, or both.

In conclusion it will be readily apparent that great progress has been made to improve the comfort of airplane passengers, and the development of the special type of heating and ventilating system for Boeing Stratoliners marks another milestone in the progress of providing comfort for high altitude flying.

Come to Pacific Coast Meeting and Exposition in June

THE Pacific Coast members are making elaborate preparations for the Society's Semi-Annual Meeting 1941, to be held at the Palace Hotel, June 16 to 19. Neil H. Peterson, President of the Golden Gate Chapter, has announced the personnel of the Committee on Arrangements with G. M. Simonson as General Chairman and Clyde Bentley as Vice-Chairman. A fine technical program is being planned by the Meetings Committee and there will be interesting sightseeing and inspection trips to entertain visiting members.

The Pacific Coast Meeting offers an opportunity for discussion of technical problems and it can be combined with a fine vacation for the entire family. Through cooperation with the other Pacific Coast Chapters a series of attractive trips is being arranged in Los Angeles, Portland and Seattle.

Transportation arrangements contemplate the assembly of members from eastern districts at Chicago, from which point they can travel in a group to San Francisco. With various optional return routes it is possible to develop a trip that will meet every requirement of time, budget and scenic beauty.

It is time for members to plan for this Pacific Coast meeting and detailed transportation information can be supplied by local travel offices.

The meeting this summer will have an added feature in the Pacific Heating and Air Conditioning Exposition, which will be held during the entire week of June 16 at the Civic Auditorium in San Francisco. Advance space reservations indicate a fine exhibit of equipment devoted to heating and air conditioning which will be of interest to members of the profession and industry as well as to the public.

Leaders in the contracting field will be in San Francisco for the Annual Convention of the *Heating, Piping and Air Conditioning Contractors National Association* at the St. Francis Hotel during the week of June 16. This combined program for the heating, ventilating and air conditioning profession and industry will focus attention on San Francisco during the week of June 16, 1941.

The Society's Pacific Coast Members are unanimous in sending a cordial invitation to their fellow members to come to California in June. This message was delivered to the Society at its 47th Annual Meeting in Kansas City by Neil H. Peterson, President of the Golden Gate Chapter, and was heartily seconded by the cooperating Pacific Coast Chapters' representatives.

WESTERN MICHIGAN HOST TO W. L. FLEISHER

December 17, 1940. The Western Michigan Chapter held its monthly meeting at the Rowe Hotel, Grand Rapids, with an attendance of 55 members and guests. Pres. T. D. Stafford called the meeting to order followed by the reading of the minutes of the previous meeting.

Prof. L. G. Miller was called upon to present the Past President's plaque to Pete Wierenga, which was done in a very suitable manner, and much to everyone's surprise Mr. Wierenga was able to respond.

Following the announcement of the January meeting, President Stafford introduced the guest of the evening, W. L. Fleisher, consulting engineer of New York, and First Vice-President of the ASHVE, who gave a very interesting talk on bomb shelters as designed and built in England. He explained that these shelters had certain minimum ventilation requirements and it was brought out that the British authorities had to come to this country to obtain data on what actually was needed in the way of ventilation. Mr. Fleisher used numerous slides to illustrate his talk, which was followed by considerable discussion of interest to the members. According to Secretary Warren's report the meeting adjourned at 10:00 p.m.

MINNESOTA HEARS DISCUSSION ON ELECTROSTATIC AIR CLEANERS

December 2, 1940. The December meeting of the Minnesota Chapter was held at the Coffman Memorial Union on the University of Minnesota Campus, with 43 members and guests present. Dinner was served preceding the meeting, which was called to order by Pres. M. H. Bjerken, and the minutes of the October meeting and a report on the November party were read and approved.

R. E. Backstrom discussed the progress of the alumni committee in connection with the proposed mechanical engineering building at the University and urged the support of the Society's Minnesota Chapter membership in this work. At the close of Mr. Backstrom's remarks it was voted that the Minnesota Chapter direct a letter to the Senate Finance and House Appropriation Committees in support of this project.

Secretary Anderson read a letter from the Secretary of the ASHVE, advising the Minnesota Chapter that a chapter delegate and representative on the Nominating Committee to function at the Annual Meeting in Kansas City, should be selected before January 1. It was voted that the Executive Committee make this selection.

The speaker of the evening, H. H. Van Epps, Westinghouse Electric and Mfg. Co., presented an interesting discussion of electrostatic air cleaners. According to the report of the secretary, the meeting adjourned at 8:45 p.m.

WISCONSIN CHAPTER HOLDS JOINT MEETING WITH ASRE

December 19, 1940. The 4th meeting of the fall season of the Wisconsin Chapter was held at the City Club in joint session with the *American Society of Refrigerating Engineers,* with a total of 49 members and guests present.

The reading of the minutes of the previous meeting was suspended, and Pres. A. S. Krenz then introduced Mr. Panlener of the *ASRE* group, who outlined briefly the subject of the next program. The next joint meeting is to be sponsored by the *ASRE.*

The secretary was then directed to read a copy of the resolution, prepared by the Professional Engineers committee, so that the recommendation of the committee could be discussed in open meeting. Messrs. Randolph, Frentzel, Krenz, C. W. Miller, Ouweneel, Goldsmith, Nelson and Szekely presented their views. Mr. C. W. Miller then made a motion to delete all of the report with the exception of the second paragraph and to present it to the Industrial Commission substantially that way. This motion was amended by Mr. Randolph and seconded by Mr. Frentzel to the effect that a recommendation as outlined be referred back to the committee for further discussion and then presented in open meeting at a future date, and the motion was carried.

President Krenz then introduced the main speaker of the evening, F. C. Houghten, director of the ASHVE Research Laboratory, Pittsburgh, who gave a very interesting talk on solar effect on building structure, including glass block and data on sprayed and flooded roofs, which he illustrated with slides. The meeting was then opened to discussion and questions from the floor were cheerfully answered by Mr. Houghten. After a rising vote of thanks the meeting was adjourned, according to Secretary Schreiber's report.

PITTSBURGH ELECTS 1941 OFFICERS

December 9, 1940. The annual meeting of the Pittsburgh Chapter was held at the Roosevelt Hotel with 25 members and guests present.

Before proceeding with the regular business, Pres. F. C. McIntosh introduced Dr. N. A. Cleven, professor of history at the University of Pittsburgh, who spoke on International Relations. The subject was timely and interesting and a lengthy discussion followed the close of Dr. Cleven's address.

The business session opened with the reading of the minutes of the November meeting which were approved as read. J. F. Collins, Jr., member of the Society's Council, reported on recent actions of the Council which affect chapter operations.

President McIntosh gave a brief annual report of his term as chapter president, in which he suggested the changing of the date of the Chapter's annual meeting from December to May. Following considerable discussion from the floor, it was moved by Mr. Collins and seconded by J. L. McCullough, that the secretary draft a proposed amendment to the By-Laws and mail a copy of it to the members with the January meeting announcement, and the motion was passed.

P. A. Edwards, chairman of the

nominating committee, reread the committee's report, and as there were no further nominations from the floor, it was moved by J. J. Tennant and seconded by H. Lee Moore, that the secretary cast a unanimous ballot for the candidates as nominated, which resulted in the following officers being elected for the year 1941:

President—E. C. Smyers
Vice-President—C. M. Humphreys
Secretary—T. F. Rockwell
Treasurer—L. S. Maehling
Board of Governors—F. C. McIntosh, *chairman;* G. G. Waters and H. B. Steggall.

Retiring President McIntosh then turned the gavel over to President Smyers who gave a brief acceptance speech and then adjourned the meeting at 10:45 p.m., according to Secretary Rockwell's report.

MEETINGS HELD IN ST. LOUIS

December 3, 1940. The regular monthly meeting of the St. Louis Chapter of the Society was held at the Kingsway Hotel, and the minutes of the November meeting were approved.

Following the report by the treasurer, the secretary read a letter from the Kansas City Chapter suggesting that one or more railroad coaches be made up especially for group attendance of Society members leaving St. Louis for the Society's 47th Annual Meeting in Kansas City. Secretary Boester then made inquiry to determine how many of those present planned to go to this meeting, and what number were going by rail. Since a good many planned to go by rail, Mr. Boester was instructed to arrange with the railroad for one more car to be set aside for this purpose and to see that all members and those interested in attending the Annual Meeting were advised of the rail fare and schedules.

J. H. Carter, chairman of the program committee, announced plans for coming meetings, including that of March 4, at which William Goodman, La Crosse, Wis., will speak.

B. C. Simons and E. E. Carlson then presented a discussion of the paper entitled Some Factors in Air Conditioning Design for a Federal Office Building, by W. A. Brown (published in ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning,* November, 1940). This was exceptionally well given and equally well received.

After a brief recess, R. B. Cleveland, sales engineer of the Acoustical Dept., Johns-Manville, St. Louis, who was the principal speaker of the evening, gave an instructive discussion on the subject of the theory of acoustics. Following Mr. Cleveland's talk the meeting adjourned at 9:30 p.m.

November 12, 1940: The November meeting of the Chapter was also held at the Kingsway Hotel, and 27 members and guests were in attendance. The minutes for the October meeting were approved as read.

J. S. Rosebrough, chairman of the membership committee, gave his report, and new booklets describing the accomplishments and activities of the Society and containing information on membership, were distributed to those present. Pres. C. E. Hartwein offered several suggestions for securing new members and it was agreed to write all St. Louis members of the Society who had not already joined the Chapter, inviting them to join.

President Hartwein then presented new Rules and By-Laws, reconstructed by R. J. Tenkonohy, for further discussion, and a vote of their adoption to replace the present By-Laws. There being no discussion, Mr. Carlson offered a motion they be accepted, which was seconded by L. R. Szombathy, and the motion was carried unanimously. President Hartwein then thanked Mr. Tenkonohy for the time and effort required to prepare these new Rules and By-Laws.

Mr. Carter, chairman of the meetings committee, suggested a question program for the January meeting instead of a speaker.

W. P. Norris presided during the meeting program, at which time there was a discussion, led by L. J. DuBois on the paper Summer Cooling Load as Affected by Heat Gain Through Dry, Sprinkled and Water Covered Roofs, by F. C. Houghten, H. T. Olson and Carl Gutberlet (published in ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning,* July, 1940). This discussion was received with much interest and many questions were asked.

Mr. Norris then introduced the principal speaker of the evening, G. C. Early, Jr., formerly assistant manager of air conditioning research, *American Association of Railroads,* who addressed the meeting on railway car air conditioning.

CONVECTORS AND CAST-IRON RADIATION SUBJECT AT PHILADELPHIA

December 12, 1940. The regular meeting of the Philadelphia Chapter was held at the Engineers' Club, and was called to order by Pres. C. B. Eastman.

Brig. General Owen of the Salvation Army spoke for a few minutes of the work of the Salvation Army and requested support of the annual drive. This was followed by a brief talk by F. G. Thomas, Jr., who spoke in support of the annual campaign for the United Charities.

The minutes of the November meeting were read and approved, followed by the treasurer's report, which was accepted as read.

President Eastman advised the members of a meeting of the *Association for Advancement of Science,* and then called attention to President Giesecke's letter requesting information for the service records of members of the Society.

Merrill F. Blankin was called upon to comment on the Society's 47th Annual Meeting in Kansas City, and it was brought out that a railroad was providing special cars for those going to Kansas City.

H. H. Mather, chairman of the meetings committee, announced that all arrangements were made for the annual meeting in January and urged the members to make their reservations early.

H. H. Erickson introduced the speaker of the evening, T. A. Novotney, Johnstown, Pa., who addressed the gathering on the subject of convectors and cast-iron radiation. Mr. Novotney explained the fundamental factor of design which affects the results secured in convectors and cast-iron radiators.

His talk was enthusiastically received and considerable discussion followed the presentation. As reported by Secretary Caldwell, the speaker was given a rising vote of thanks, and the meeting was adjourned at 9:20 p.m.

NOISES IN VENTILATING AND AIR CONDITIONING SYSTEMS DISCUSSED AT MASSACHUSETTS

November 19, 1940. The regular monthly meeting of the Massachusetts Chapter was attended by 53 members and guests. Following a poll on licensing of engineers, which resulted in 4 in favor and 18 against, Mr. Hagen delivered an address on the noises in ventilating and air conditioning systems. He stated that the noises were largely confined to fan noises and method of control, and classified fan noises into swishing or whistling noises, and low, irregular rumbling noises. The former of these he went on to describe as high-frequency noises penetrating and very unpleasant, but which may be absorbed readily by any commercial sound-absorbing material. The latter he defined as low-frequency noises, due to the action of fan blades on air. These are often mistaken for mechanical noises, he explained, and a low rumble occurs frequently in unoccupied buildings and is usually *masked* by normal activities of persons when the building is occupied. Mr. Hagen further stated that with fan static pressures less than 1½ in. there is usually little noise to require acoustical treatment, and trouble may arise when static pressures exceed 1½ in. He explained that special building conditions make a great difference in the noise, and it is important that the size of sound-absorbing squares be one-half the wave length of the sound to be absorbed.

Mr. Hagen deplored the lack of a standard code for measuring noise, and stated that in his opinion merely specifying the decibel level of fans is not satisfactory, and further explained that sound characteristics of a room, distance from the fan at which noise is measured, sound frequencies, etc., should be specified or standardized. In conclusion Mr. Hagen brought out the fact that still less is known about permissible total noise levels in rooms, and all that can be said at present is that an increase of 3 db above the normal room noise level might be acceptable for most practical purposes.

STEEL BOILERS TOPIC AT MANITOBA MEETING

November 21, 1940. The monthly meeting of the Manitoba Chapter of the Society was held at the Fort Garry Hotel, Winnipeg, with 14 members and guests in attendance. Pres. P. L. Charles called the meeting to order, and the minutes of the October meeting were read and approved.

A motion was proposed and duly carried that Messrs. Charles, Kent and McDonald use their own discretion in selecting a suitable remembrance to be forwarded to Pilot Officer P. H. Foster now in Scotland.

The secretary read a letter received from New York relative to educational films and was instructed to refer this letter to the speakers committee chairman, D. F. Michie. The secretary then submitted his report, followed by a report by the membership chairman, William Glass.

J. R. Stephenson introduced the speaker of the evening, E. Anderson, who gave a very interesting talk on the subject of steel boilers. F. L. Chester, in behalf of the Manitoba Chapter, expressed its appreciation to Mr. Anderson, and the meeting adjourned at 10:00 p. m.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 32 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by February 15, 1941, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ANDERSON, EDWIN L., Asst. Industrial Hygienist, State Health Dept., Hartford, Conn. (*Reinstatement*)	R. C. Stratton (*AICE*) J. B. Ficklen (*AICE*)	W. P. Durland (*ASME*) M. S. Smith
ARMSTRONG, CLYDE C., Dist. Engr., Frigidaire Div., General Motors Sales Corp., Des Moines, Ia.	R. A. Norman B. E. Landes	E. H. Borg N. B. Delavan
CLARKE, JOHN H., Marine Engr., Maritime Commission, Washington, D. C.	J. W. Markert F. E. Spurney	T. H. Urdahl P. H. Loughran, Jr.
CROFT, HUBER O., Prof. & Head of Mech. Engrg., University of Iowa, Iowa City, Ia.	R. A. Norman F. E. Triggs	N. B. Delavan A. L. Walters
CURL, ROBERT S., Mgr., Air Cond. Dept., Carle Electric Const. Co., Akron, Ohio.	L. E. Slawson R. T. Southmayd	D. L. Taze L. O. Weldy
FLEMING, PAUL B., Consulting Engr., Cleveland, Ohio.	L. E. Slawson D. L. Taze	W. M. Rowe L. O. Weldy